# Factors affecting the environmental performance of a naturally ventilated lecture theatre

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"When you can measure what you are speaking about, and express it in numbers, you know something about it; but when you cannot measure it, when you cannot express it in numbers, your knowledge is of a meagre and unsatisfactory kind: it may be the beginning of knowledge, but you have scarcely, in your thoughts, advanced to the stage of *science*"

Lord Kelvin (William Thomson) [1824-1907], *Popular letters and addresses* [1891-1894]

# I Abstract

A program of work involving the measurement of ventilation rates, air velocities and temperatures has been completed within the Queens Building (the Engineering and Manufacture Building, De Montfort University), and some results have been obtained.

Measurements of the above parameters have been recorded for "summer", "winter" and "mid-season" conditions, using average occupancy heat gains. Results obtained indicate that ventilation rates through one of the theatres are driven by both stack and wind induced effects, and can be much higher than predicted by computer and physical models.

Naturally ventilated auditoria are very unusual, as mechanical ventilation and/or air conditioning is usually required to deal with the relatively high internal and very variable heat gains. Due to the uniqueness of such spaces and the difficulties of performing experiments, measurements of the internal environment have never been carried out before in a naturally ventilated auditorium.

Room air temperatures show little variation with time, due to the heavy weight nature of the building structure and the large exposed areas of ceiling slab available for radiant cooling. The distribution of air into the auditorium was found to be non-uniform due to the asymmetric layout of external inlets. High inlet air velocities (greater than 0.5 m/s) and low air temperatures (less than 10 °C) were found when openings were fully open for outside temperatures of 5 °C or less.

The results indicate the beneficial effect of the thermal mass of the building and the night venting regime used.

Various improvements can be made to the ventilation system as designed. Suggested modifications include the addition of blanking plates over inlet grilles on the right hand side of the theatre, the installation of manual control and the reduction of the lowest inlet setting area (preventing large inlet velocities in winter).

# I Abstract (continued)

Empirical models of the environmental performance of the theatre has been developed. These allow internal temperatures and flow rates to be predicted for a variety of external temperatures, and internal heat gains. The use of a ventilation model is demonstrated which predicts flowrates for particular wind speeds and directions.

# II Acknowledgements

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# VI Notation

| | |
|---|---|
| $A$ | area ($m^2$) |
| $A_E$ | Effective area ($m^2$) |
| ACR | air change rate ($h^{-1}$) |
| $C_d$ | discharge coefficient |
| $C_e$ | external concentration of tracer gas (ppm) |
| $C(t)$ | internal concentration of tracer gas at time t (ppm) |
| $c_p$ | specific heat capacity (kJ $kg^{-1}$ $K^{-1}$) |
| D | characteristic dimension (m) |
| DP | pressure difference (Pa) |
| DT | temperature difference (°C) |
| $f_{ae}$ | effective area (actual) /effective area (maximum) |
| $G(s)$ | Transfer function (Laplace domain) |
| $h_1$ | height above datum of "inlet" (m) |
| $h_2$ | height above datum of "outlet" (m) |
| $I(s)$ | Input (Laplace domain) |
| $k_{os}$ | flow coefficient ($m^3$ $s^{-1}$) |
| $n_i$ | flow exponent |
| $N_{occ}$ | number of occupants |
| $n_{pts}$ | number of points |
| $O(s)$ | Output (Laplace domain) |
| $\Delta P$ | Pressure difference (Pa) |
| $P_{wi}$ | wind pressure at a point i (Pa) |
| $q_v$ | volumetric flowrate ($m^3$ $s^{-1}$) |
| $q_{vB}$ | volumetric (buoyancy driven) flowrate ($m^3$ $s^{-1}$) |
| r | room |
| $\rho$ | density (kg $m^{-3}$) |
| $\rho_c$ | density of outside air (kg $m^{-3}$) |
| $\rho_h$ | density of internal air (kg $m^{-3}$) |
| $s_1$, $s_2$ etc. | various outlet configurations |
| $T_{ai}$ | internal air temperature (°C) |

# VI Notation (continued)

| | |
|---|---|
| $T_{ao}$ | external air temperature (°C) |
| $T_{avg}$ | average temperature (°C) |
| $T_{base}$ | average external temperature (°C); |
| $T_c$ | ceiling surface temperature (°C) |
| $\tau_c$ | time constant (h; cooling) |
| $\tau_h$ | time constant (h; heating) |
| $T_{stk(lhs)}$ | stack air temperature (°C); left hand side |
| $T_{stk(rhs)}$ | stack air temperature (°C); right hand side |
| $T_{var}$ | variation of external temperature |
| $\mu$ | dynamic viscosity ( Ns $m^{-2}$) |
| $v_{avg}$ | average room air speed (m $s^{-1}$) |
| $v_{fs}$ | air velocity (m $s^{-1}$) (full scale) |
| $v_m$ | air velocity (m $s^{-1}$) (model) |
| $v_w$ | wind speed (m $s^{-1}$) |
| $\omega$ | angular frequency (rad $s^{-1}$) |

# VII Abbreviations

| | |
|---|---|
| avg | average |
| BMS | building management system |
| BRE | building research establishment |
| c. | circa (approximately) |
| calc. | calculated |
| CFD | computational fluid dynamics |
| CIBSE | chartered institution of building services engineers |
| clo | clothing insulation value ($m^2 K W^{-1}$) |
| $CO_2$ | carbon dioxide (tracer gas) |
| CON(i) | concentration (of gas) at point i |
| dir'n | direction |
| DMU | De Montfort University |
| EMA | East Midlands Airport |
| ESP | dynamic thermal analysis software |
| fan (on/off) | stack fan status (1 or 0) |
| hl | high level |
| htg v | heating valve (relative position, %) |
| htg sys | heating system |
| IAQ | indoor air quality |
| IEA | international energy agency |
| lhs | left hand side |
| met | metabolic rate ($W m^{-2}$) |
| N, S, E, W | north, south, east, west |
| PC | personal computer |
| pft | perfluorocarbon tracer |
| PMV | predicted mean vote |
| PPD | percentage of people dissatisfied (%) |
| PSV | particle streak velocimetry |
| ppm | parts per million (v/v) |
| RH | relative humidity (%) |

# VII Abbreviations (continued)

| | |
|---|---|
| rhs | right hand side |
| $SF_6$ | sulphur hexafluoride (tracer gas) |
| stk | stack |
| 3-D | three dimensional |
| V | effective volume ($m^3$) |
| vel | velocity ($m\ s^{-1}$) |
| vent pos'n | opening vent position (%) |
| wind sp | wind speed ($m\ s^{-1}$) |
| vroom | room air velocity ($m\ s^{-1}$) |
| vstk(lhs) | Stack air speed ($m\ s^{-1}$); left hand side stack |
| vstk(rhs) | Stack air speed ($m\ s^{-1}$); right hand side stack |

# 1 Introduction

## 1.1 Research objectives

The main objectives of this research project were to:

- Assess the ventilation performance of a naturally ventilated auditorium using appropriate experimental techniques.
- Develop an empirical model of the ventilation system used. This would allow environments within enclosures with similar thermal masses and ventilation openings to be simulated.
- Assess the thermal comfort conditions within the space.

## 1.2 Literature review - overview

The literature review consists of the following sections:

(i) a review of the design and operation of non-domestic naturally ventilated buildings;
(chapter 1; sections 1.3 to 1.4)

(ii) a summary of computer modelling and physical modelling techniques;
(e.g. the use of dynamic thermal (computer) models; salt bath modelling)
(chapter 1; sections 1.5 to 1.6)

(iii) a detailed discussion of the various measurement methods for measuring air flowrates, ventilation effectiveness, air speeds, temperatures, and techniques for air flow visualisation in naturally ventilated spaces. This allows the various techniques to be examined in depth and appropriate methods to be selected for the auditorium.
(chapter 2; sections 2.1 to 2.8)

The literature review indicated that very few naturally ventilated auditoria have been constructed and that no experimental monitoring had been carried out in such enclosures.

Hence this is the first time that experimental monitoring has been performed and reported upon for a naturally ventilated lecture theatre.

## 1.3 Background : Benefits of low energy buildings

There have been moves within the last 10 years to design buildings that have low environmental impacts, i.e. buildings with low energy requirements both in terms of their needs before and during construction and over the lives of the buildings. The aim would be to produce developments which produce relatively low volumes of $CO_2$ and still maintain adequate thermal comfort levels.

The provision of services for heating, ventilating and air-conditioning of a buildings has historically been considered as work which needs to be carried out after the design of buildings structure and layout has been completed. The architect comes up with his or her grand vision of how the building should look, perhaps placing a large emphasis on providing daylighting via an extensively glazed facade. The structural engineer then

provides the design that will fulfil the architects requirements. Eventually the detailed design will be passed on to the building services engineer who is told "here's the building, now design the air conditioning system". The design process thus has been sequential and hierarchical with information flow in one direction only. The services engineer has been programmed to provide the right number of "boxes" (i.e. boxes containing fans, pumps etc.) for a fixed design, with no regard for the energy use of the building and its likely impact on the environment.

However, a new dawn is upon us. More clients are coming to the conclusion that it makes sense to be environmentally aware. In fact, the financial rewards can be substantial, in terms of reduced capital costs and energy and maintenance costs. A university has opted to construct a naturally ventilated library as the universities requirements could be fully satisfied by such a building design and the energy savings over the projected life of the building are likely to be substantial. In addition, there may be no drop in the quality of the internal environment which results, the appearance of the building and the buildings effects on its occupants productivity levels.

How can clients be encouraged to ask for naturally ventilated buildings? How can low energy building designs be made to work in practice? The answer to the first question can be provided from surveys and detailed studies of existing "new generation" low energy buildings. These buildings do "work". Their energy use is considerably lower than more conventional buildings, they can provide year round comfort and they are easy on the eye. They can also be used to make a statement and to reflect the aspirations of an enlightened client; i.e. "Yes, we believe in saving our environment for future generations". The second question is answered by getting the building services engineer to work along-side the architect and other members of the design team at the initial stages of a project, by using brainstorming sessions to tease out key design principles, at the detailed design stage by the use of exposed thermal mass, external shading, night venting and making the building adapt to the needs of the occupants. This last point can be dealt with partly by using variable ventilation rates and adjusting lighting levels.

## 1.4 Review of literature on low energy buildings

Many references were examined to determine the current state of the art with regard to the design, construction and monitoring of low energy buildings. Very few references were found dealing with the design or monitoring of internal conditions within lecture theatres ventilated using natural ventilation; in most cases such spaces are mechanically ventilated and/or air conditioned. Auditoria generally have high internal heat gains which can be very variable; this usually dictates that an all air (conditioning) system is used to ensure thermally comfortable conditions can be maintained.

The following buildings, built before the Queens Building, have shown that adequate ventilation and thermally comfortable conditions can be provided using low energy systems. Specific references which detail the environmental performances of the buildings, are referred to in the following sections. A number of buildings which have been completed more recently, have also demonstrated the above.

### 1.4.1 Gateway 2, Basingstoke

#### (a) Building Description

Gateway 2 is a seven storey development built around a 45 m long by 22.5 m wide atrium [1]. It is located on a sloping site and is developed such that the main entrance, which is at the top of the slope, is on the third floor level. This is also the atrium floor level.

Above the atrium floor level, there are five levels to the west half of the building and four to the east, providing a clerestory level of glazing at the east end. These floors, which contain the office accommodation, are wrapped around the central rectangular atrium. Each office is generally 13.5 m deep from external wall to atrium boundary wall.

Externally the building is of an imposing black appearance. It has a stepped garden terrace, which provides a pleasant external appearance to the building. Another external feature is provided by the 1.5m deep louvred sunscreen/walkways at each floor level around the entire perimeter of the building.

Total floor area is 14,000 $m^2$ and the floor to ceiling height is 2.85 m on each level.

#### (b) Passive features.

The main passive solar feature of the building is the atrium. During the winter it provides a buffer space which reduces heat losses from offices (compared to the originally envisaged open courtyard) and a dilution volume for the stale air from the offices.

During the summer the atrium assists with the provision of natural ventilation and hence the cooling of the building. Solar gains through the atrium glazing warms the air in the atrium which induces a stack effect. When the rooflights are opened air is drawn into the atrium from the offices via the external and louvre windows.

Extensive external shading is provided which limits solar gains into occupied areas.

Exposed ceilings are used in office areas; these provide large surfaces for radiant cooling and are precooled using night venting.

### (c) Building performance

The overall energy use of the building is 194 kWh/m$^2$ which compares well with best practice figures provided by the energy efficiency office.

The average summer ventilation for the offices was found to be 4.6 ach which was lower than predicted. The winter value was 0.6 ach. Some restriction to flow is caused by perimeter offices and wind induced pressure differences can sometimes act against those generated by the stack effect.

Generally internal air temperatures in summer have been less than external temperatures. Night venting was effective in precooling ceiling slabs thus ensuring moderate dry-resultant temperatures during the daytime (dry-resultant temperature equals the average of air temperature and mean radiant temperature) [1].

### 1.4.2 Malta Brewery

The Malta Brewery is an industrial building which makes no use of cooling plant, despite the very high external summertime temperatures experienced (35 °C and above) [2]. Night-time temperatures are much lower than day time temperatures, and relatively constant internal temperatures are maintained due to the high thermal mass of the building. Chimneys are used to promote stack ventilation, which are used to provide night venting.

As for the Queens Building, the salt bath technique [5] was used to model the ventilation performance of the building. This proved to be an effective way of testing the ventilation strategy used.

Monitoring has indicated that temperatures can be maintained between 25 and 27 °C in the process halls, when the outside temperatures vary between 22 and 35 °C. Night-time ventilation is providing significant draughts and can remove 20 to 30 % of the day-time heat gains.

## 1.5 Rationale for constructing a new naturally ventilated building in Leicester

In 1989, De Montfort Universities older buildings were deemed to be "inoperable and unsafe". Following a detailed investment assessment in association with the Polytechnic and Colleges Funding Council, the decision was made to use the University as a catalyst for the regeneration of the campus and surrounding areas. A new building was to be built, this would house the school of Engineering and Manufacture.

The original brief called for an innovative design which would have minimal environmental impact and would use traditional construction techniques. The architects Short Ford Associates were selected and their initial proposal for the new building was for a naturally ventilated, daylit building.

Usually a sealed mechanically ventilated or air-conditioned building would be required for such an urban location, due to high ambient noise and pollution levels, and high internal heat gains. However, the success of the architects design for a process building at Farsons Brewery in Malta, which used the combination of high thermal mass and stack driven ventilation to achieve comfortable conditions convinced the architects that the same philosophy ought to work in the UK. Lower "summer" external temperatures are experienced in the UK, so it was considered likely that comfortable internal conditions would also occur in a naturally ventilated, thermally massive enclosure built in the UK.

The concept for the Queens Building was therefore a thermally massive, well insulated envelope with a limited plan depth and above average ceiling heights to promote natural ventilation and daylighting.

## 1.6 Building details

### 1.6.1 General description of the building and installed systems

The main aims of the architect Short-Ford and Partners, and design engineers Max Fordham Associates, was to design a low energy university building, which would minimise building energy and maintenance costs while still achieving acceptable ventilation rates and comfort temperatures. Thus the use of ventilation and heating plant was minimal [3].

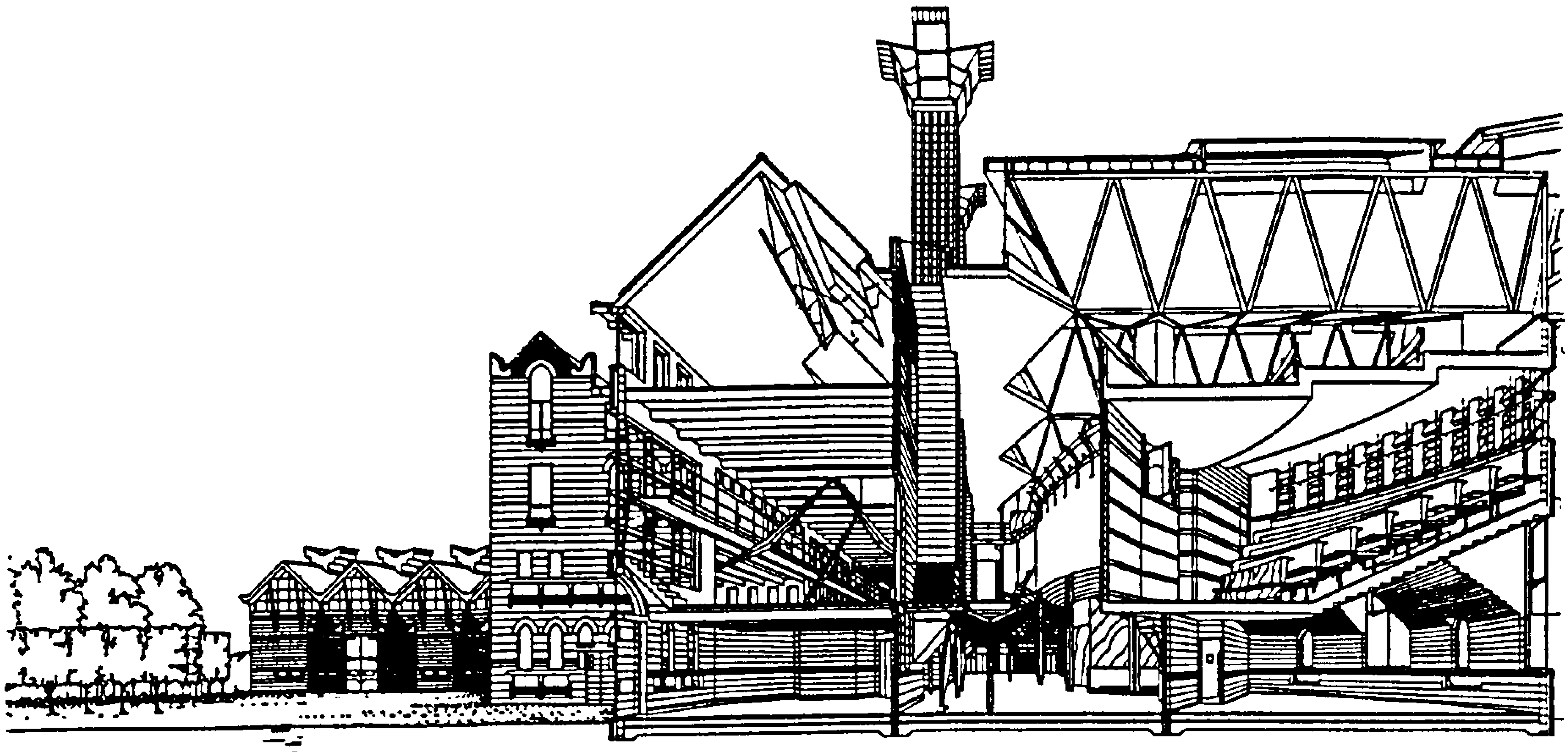
A prime feature is the use of tall ventilation stacks to promote natural buoyancy driven ventilation as a means of avoiding the use of systems which have a relatively high energy consumption, i.e. mechanical ventilation and full air conditioning. Shallow plan spaces and a large percentage of openable facade would allow effective use to be made of natural ventilation, while large areas of exposed walls and floors would increase the effects of radiant cooling. Night venting would be used in summer to pre-cool the building.

A combined heat and power plant was also installed to provide the building with its own electrical power supply and an additional source of hot water.

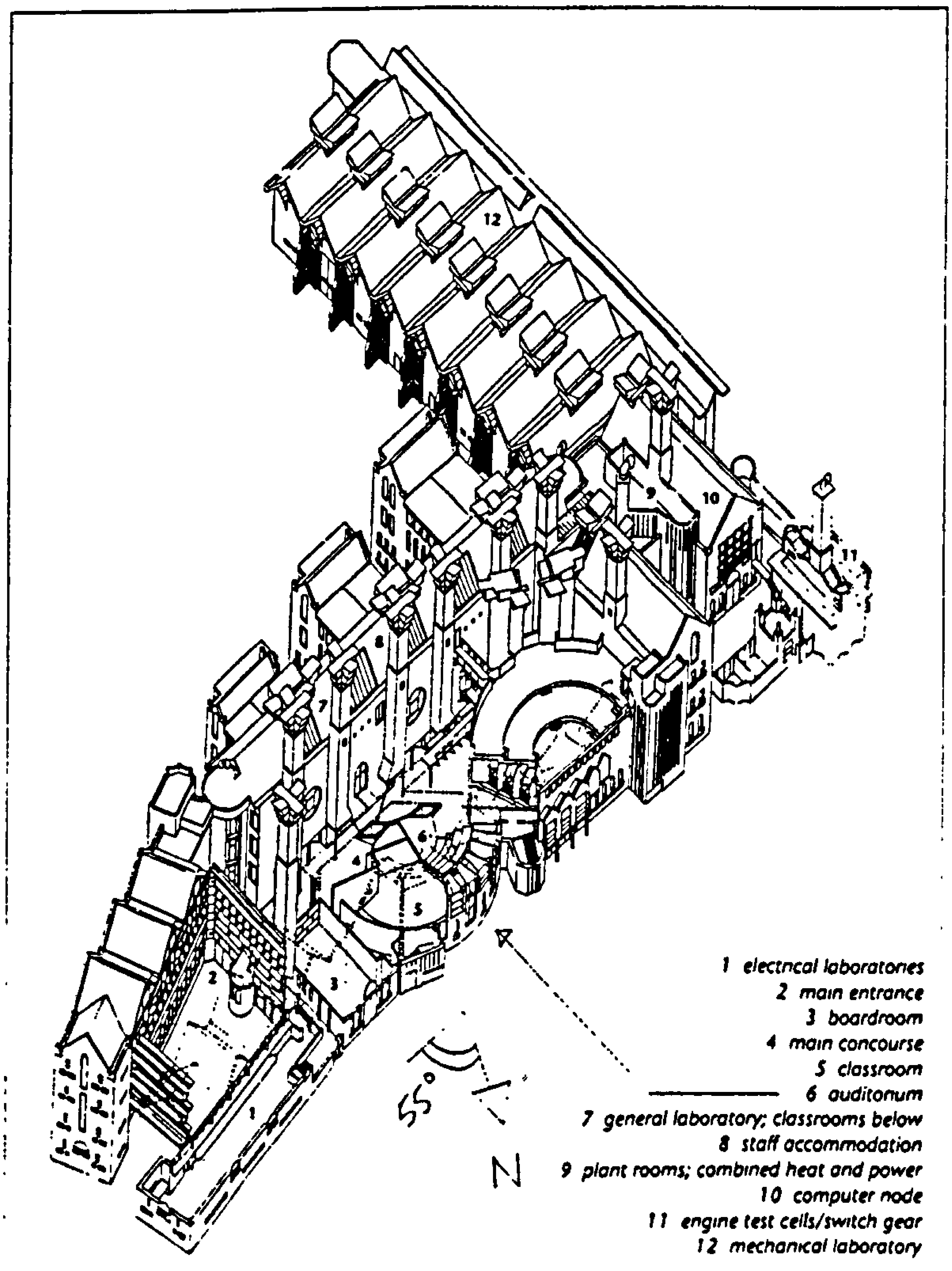
Daylighting is extensively used, this is facilitated by the use of shallow plan spaces, rooflighting and light shelves.

The building management system installed allows the ventilation and heating systems to be automatically controlled; it also records temperatures (air/water), valve/damper positions thus allowing energy use to be monitored and analysed.

The building was modelled prior to construction using a dynamic thermal analysis program and by the use of a salt bath physical model. The findings of these analyses are summarised in [4] (Eppel, Mardaljevic and Lomas) and [5] (Lane-Serff et al.) respectively.



**Above** *Perspective section.*
**Right** *Accommodation is compacted to allow cross connection between laboratories, classrooms, staff and research fellow accommodation, helping to promote cross-fertilisation between disciplines. Acoustic isolation, fire compartmentation and the natural stack ventilation paths have to be juggled with this policy.*
**Below** *The envelope is largely unframed and the construction deliberately massive in order to stabilise internal temperatures. The main Mill Lane elevation faces north west, a useful aspect for admitting controlled daylighting. Lancet windows enable glass to be pushed up into the deep tapering coffer of the precast T-beam floor structure. From left: the four-storey electronics laboratory, boardroom and school office, the north-facing gable to the drawing studios sitting above two auditoria, computer library and (extreme right) the engine testing cell.*



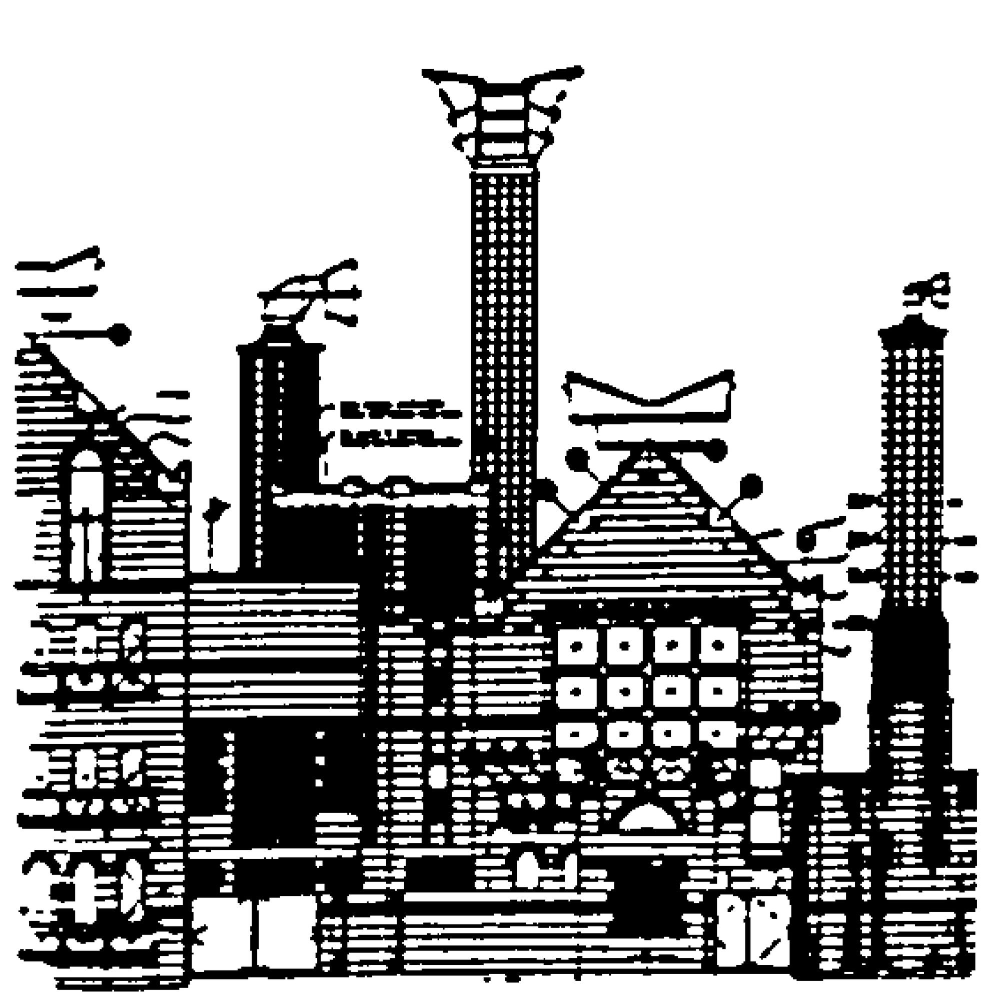



Figure 1.1 Perspective and isometric drawings of the Queens Building

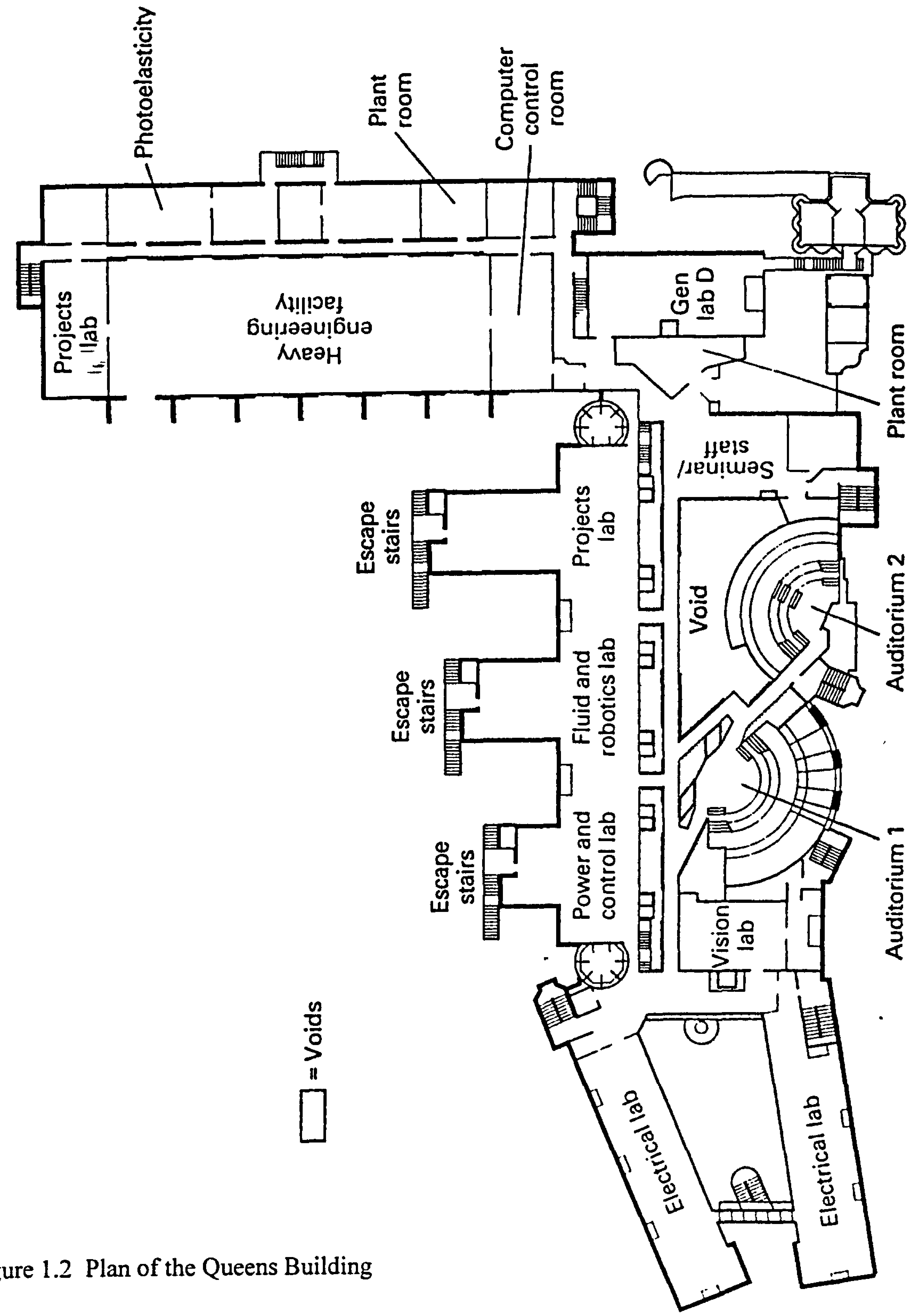


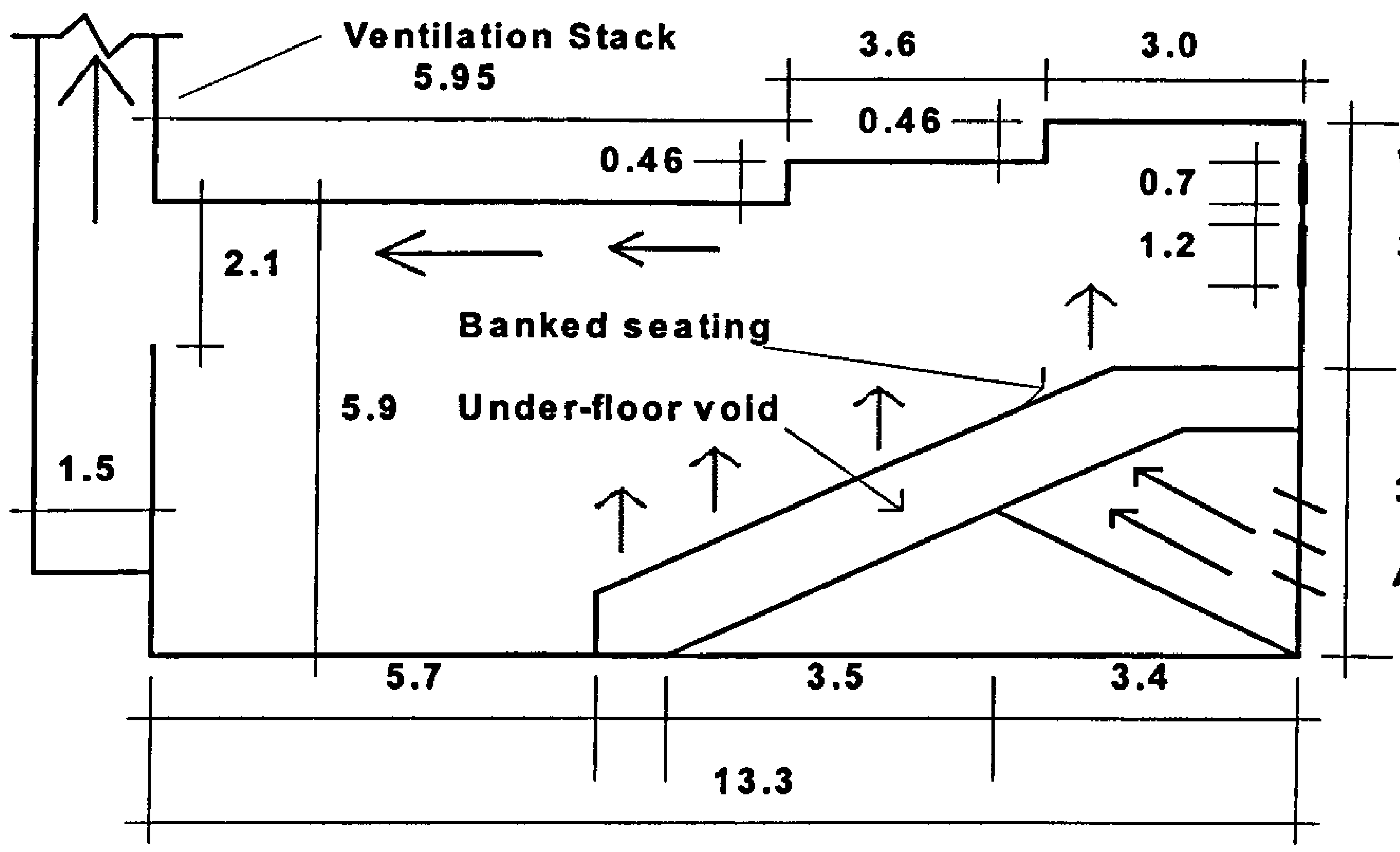

Figure 1.2 Plan of the Queens Building

The auditoria presented special problems to the appointed consulting engineers; they would have heat gains of up to 100 W/m$^2$ and stringent acoustic requirements; both very difficult to overcome without using mechanical ventilation.

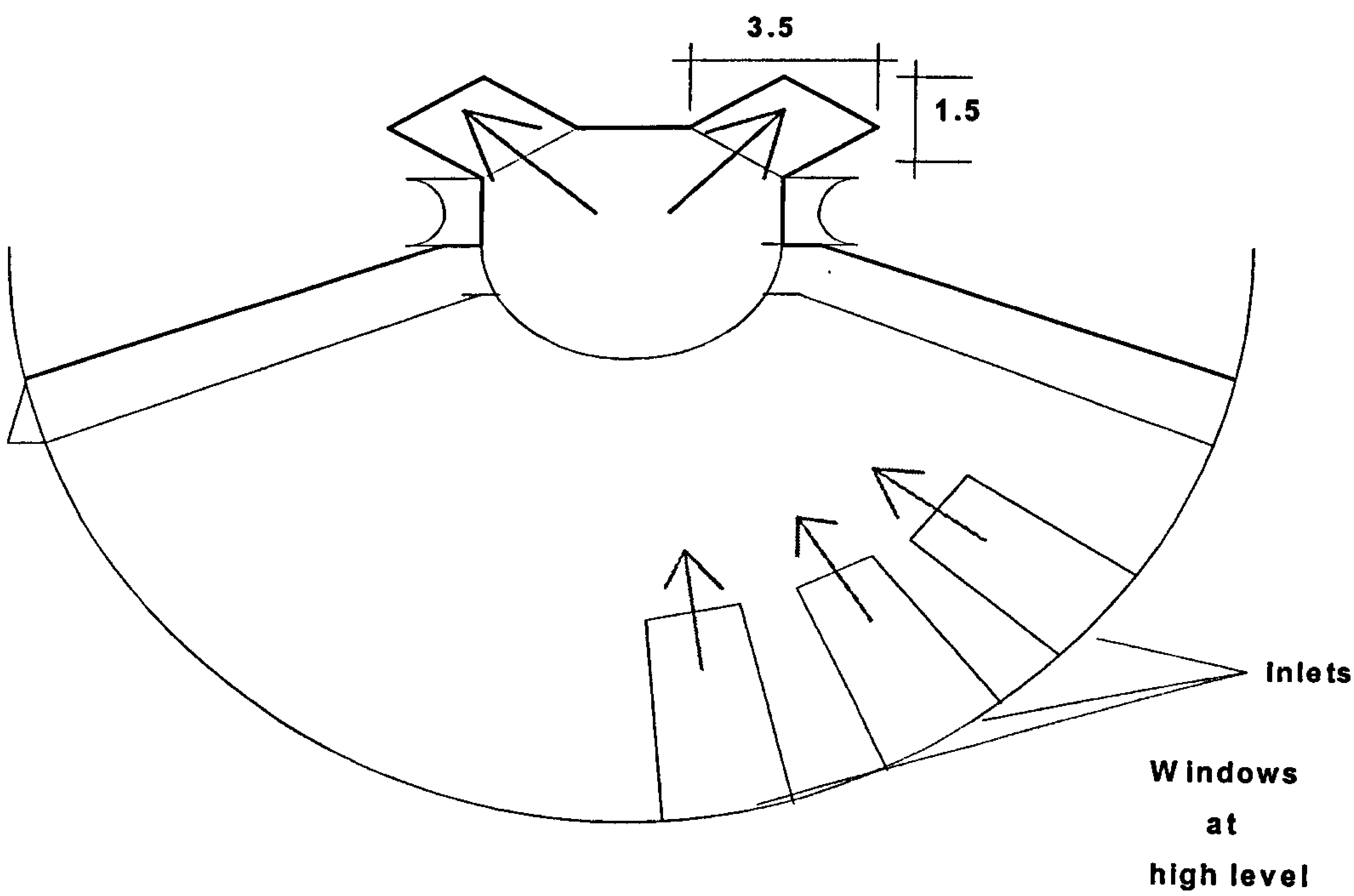
### 1.6.2 Description of Ventilation and Heating Systems

#### (a) Auditorium 1.10

The two lecture theatres have self contained natural ventilation systems. In the theatre under investigation, outside air enters adjacent to a lightly used road into three plenums via modulated control dampers (see figure 1.3), disperses within voids underneath the seating and enters the room via vertical grilles positioned at ankle height. Air leaves the space via two large openings near the base of two stacks. Outlets are positioned at the top of each stack in the form of eight windows. Inlet and outlet positions can be varied by the operation of motors controlled by the building management system (BMS). A "punka" type or "ceiling" fan is installed in one of the stacks to provide back-up ventilation in the event of inadequate stack flow rates. Heating in winter is provided by finned tubes placed within the seating void adjacent to the vertical grilles.




Section through theatre

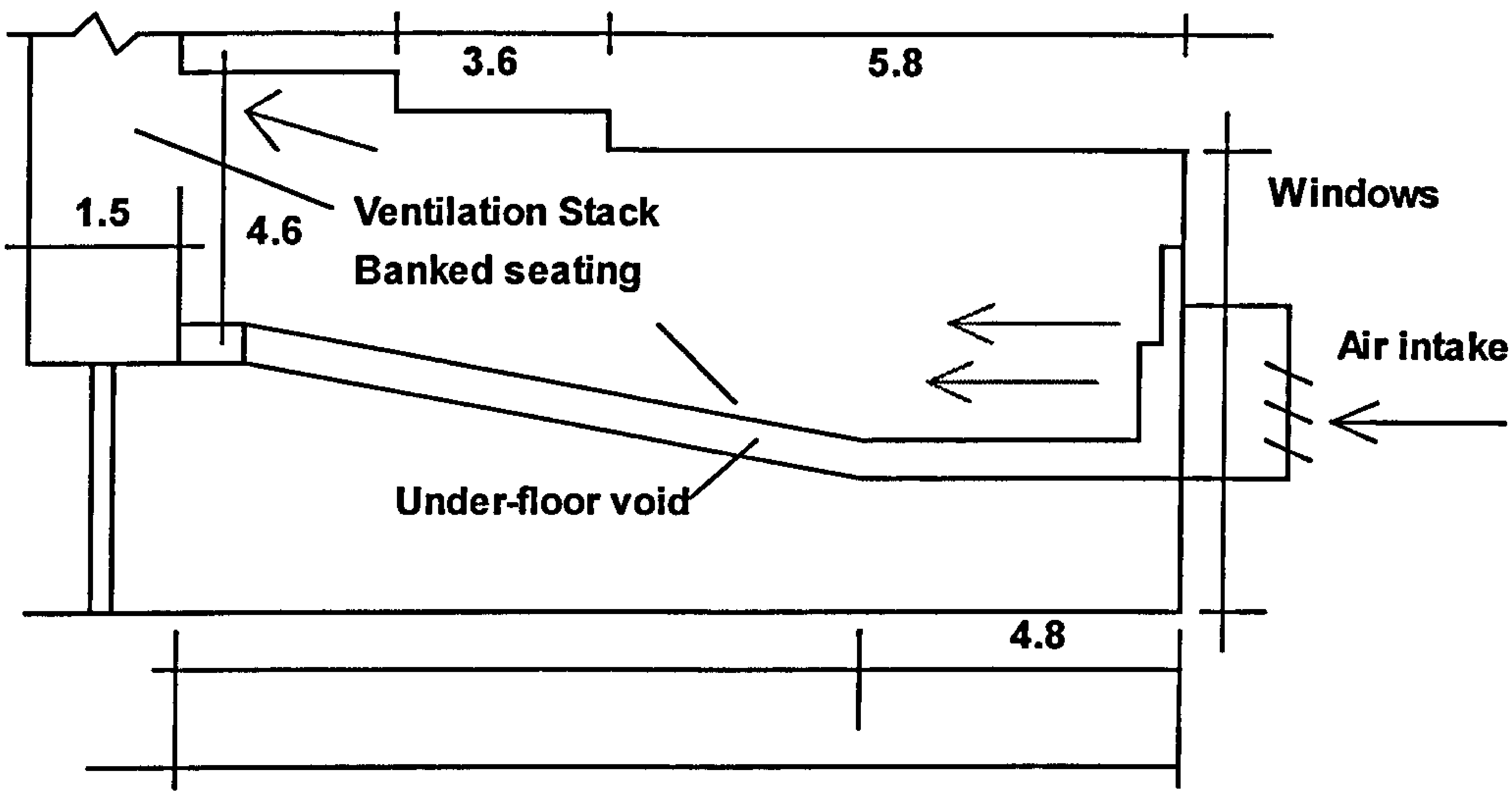



Plan(all dimensions in metres)

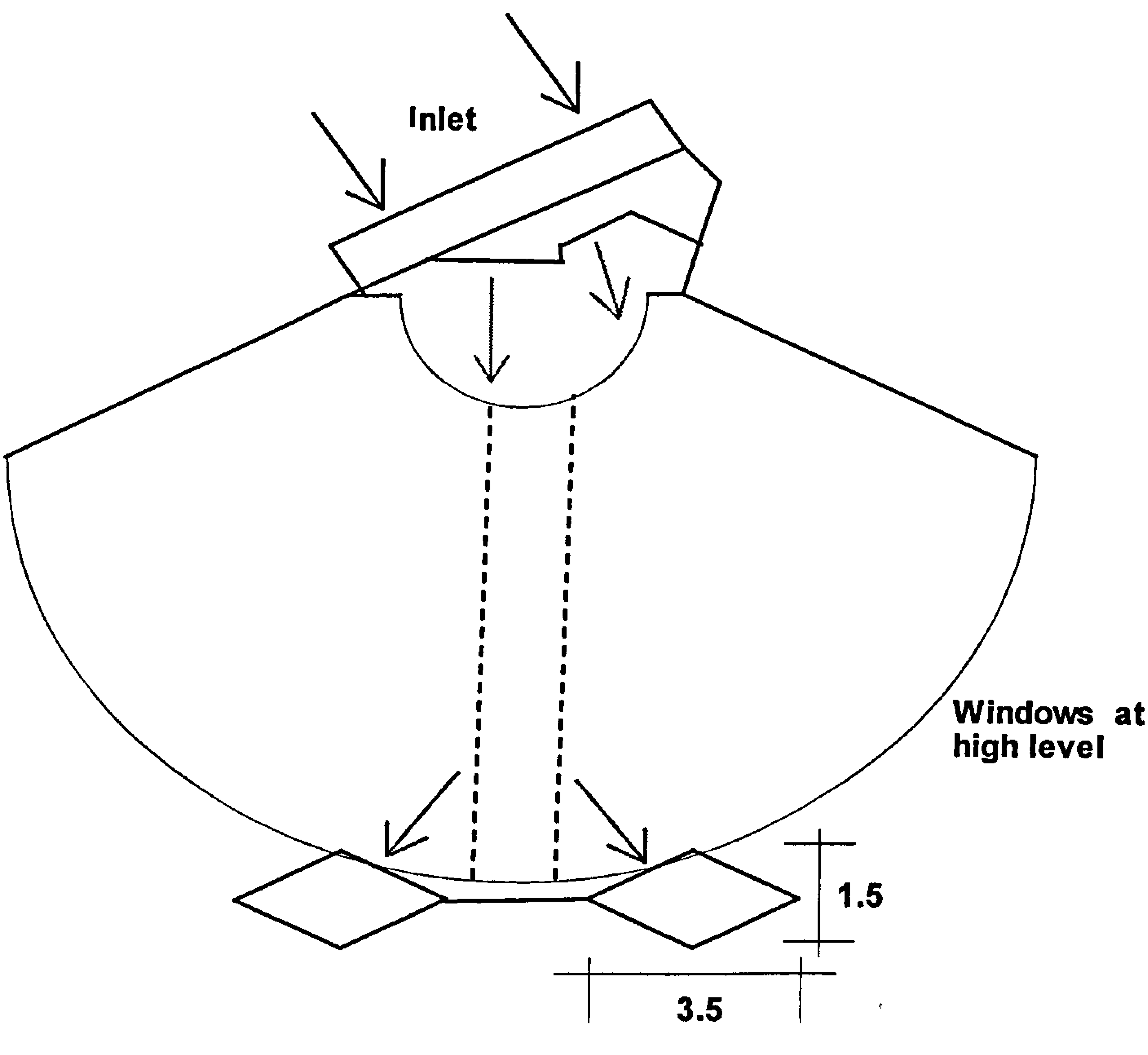
Figure 1.3 Section and plan of auditorium 1.10 showing the general directions of air movement. (Figures shown are the room dimensions).

## (b) Auditorium 1.12

The second lecture theatre has the opposite orientation of the first auditorium (see figure 1.4). Fresh air enters at the front of the space via several rows of suspended attenuators (i.e. units supported from the underfloor plenum ceilings) and two large banks of finned tubes. The design intent was that warm air would travel through the floor void and exit via the inlet grilles (located at ankle height underneath each seat). The air is then driven upwards due to combined buoyancy effects of heat transfer from finned tubes in the underfloor plenum and from occupancy heat gains. A reservoir of warmer, "polluted" air is contained beneath the ceiling. Air is exhausted via two stack openings at ceiling height and leaves via the top of the two stacks. A fan is installed in one of the stacks to initiate flows if buoyancy effects are minimal.




Section through theatre




Plan (all dimensions in metres)

Figure 1.4 Section and plan of theatre 1.12 showing the general directions of air movement (figures shown are building dimensions)

### 1.6.3 Building Management System Controls

During an occupied period, if the average internal temperature exceeds a set value (e.g. 19 °C) the inlets and outlets will open if the internal temperature is greater than the outside temperature. Otherwise the openings will close to minimum positions which are determined by the carbon dioxide sensor in the room (located in one of the stacks); i.e. 10 % open with a $CO_2$ concentration of 350 ppm and 100 % open when the $CO_2$ concentration is equal to or greater than 1000 ppm [6].

A mixing valve plus actuator varies the flow rate of hot water to the heating circuit and is controlled using the average room temperature.

The fans (one in each theatre) are started if the average internal temperature is three degrees Celsius above the room set point and greater than the external temperature plus two degrees (provided the inlets/outlets are open and if it is not raining). When the space is unoccupied, in winter the openings close, and in summer they open at night to pre-cool the room to a set temperature (summer is assumed to occur when the minimum outside temperature is greater than 18 °C between the times of 16:00 and 16:10).

The simultaneous occurrence of rain and wind speeds above 25 mph (11.2 m/s) and a damper demand of greater than 50 % causes the outlets to be limited to 50 % open. If wind speeds are in excess of 40 mph (17.9 m/s) , then the openings close.

In the event of a fire, to facilitate the removal of smoke, the room inlets and stack outlets fully open and the extract fan is switched on. External air (not internal) humidity is monitored and recorded, but is not used by the control system.

## 1.7 Computer modelling of auditorium carried out by research staff at De Monfort University [4]

A check on summertime temperatures within auditorium 1.10 was required. Very large occupant heat gains could occur (up to 15 kW), combining this with lighting heat gains (2.8 kW) and a machine load of 0.5 kW would give heat gains approaching 100 W/m$^2$.

The thermal comfort conditions would be dictated by the convective and radiative heat from people and lights and the stack driven air flowrates induced by the difference in height between inlets and exhausts and the temperature difference between the internal and external air temperatures.

These time varying interactions could only be adequately explored using a dynamic thermal computer simulation program capable of simultaneously solving both the equations governing heat transfer and airflows. ESP was such a program, and it was used for the analyses.

Nine cases were examined in detail and compared to a base case. In terms of the peak dry resultant temperature, the following points can be made:

i) The dry resultant temperature would be significantly reduced during days of reduced occupancy. The reduction would be 1.3 °C at 60 % of the design value.

ii) Opening windows had the biggest effect of any one variant; a 5 m$^2$ opening reduced the peak dry resultant temperature by 0.8 °C.

iii) If both the plenum inlet and the stack cross-sectional areas were increased, then the peak dry resultant temperature would be reduced by 1.2 °C. Enlarging the chimney produced the greater effect of these two changes.

iv) Only relatively small effects were produced by heating the chimney and by night time venting ( reductions of 0.5 and 0.1 °C respectively). Mechanical venting of the room at night to cool exposed surfaces also had little effect on the peak dry resultant temperatures.

v) Acoustic tiles placed around the back third of the theatre increased the peak dry resultant temperature by 0.8 °C. This value is an average for the room as a whole, and larger values would occur in areas near to the tiles, leading to greater discomfort towards the rear of the auditorium.

The simulation program, ESP, predicted that, for the base case with acoustic tiles, there would only be 9 hours per year with dry resultant temperatures above 27 °C, and none of these would occur during a normal academic term. The building would be sparsely occupied when the highest external (summer) temperatures would occur. A 50 % reduction in the number of hours occurred when summertime overheating takes place (these occurred when the building would normally be occupied), by increasing inlet and outlet areas. Adding a openable window in the external wall at the rear of the space reduced this by a further 20 %.

The results of these analyses gave further weight to the argument that a natural ventilation strategy would work, in spite of the theatres high heat gains and stringent acoustic requirements.

## 1.8 Physical Modelling completed by Cambridge Environmental Research Consultants Ltd [5]

A physical model of the De Montfort Building was used to test whether natural ventilation could provide adequate flow rates to maintain comfortable conditions in the building. This comprised a 1:75 scale model made of clear perspex, suspended upside down in a bath of water. The model would represent conditions occurring during the summer. Heat sources are simulated by salt solutions; the falling of these solutions will model the rising of warm air above the heat sources in the real building. The salt water is dyed to make its movement visible. A side elevation of the model in the tank was filmed using an inverted video camera.

Samples of fluid were taken during experiments and the densities of these samples measured. The corresponding temperatures in the actual building were calculated from the densities of the salt solutions in the model by the use of appropriate scaling factors.

The dimensions, times, velocities and densities will differ between the model and the real building, since the same Reynolds number has been used. The Reynolds number equals:

$$Re = \frac{\rho v D}{\mu} \qquad (1)$$

where $\rho$ = fluid density;

v = fluid velocity,

D = characteristic dimension;

$\mu$ = dynamic viscosity).

Air speeds were c. 5 times bigger in the model than in the real building (i.e.

$$\frac{\rho_m v_m D_m}{\mu_m} = \frac{\rho_{fs} v_{fs} D_{fs}}{\mu_{sf}}; \qquad (2)$$

$$\text{thus } v_m = \frac{\rho_{fs} v_{fs} D_{fs} \mu_m}{\rho_m D_m \mu_{sf}}; \qquad (3)$$

$$v_m = \frac{1.2 * 75 * v_{fs} * 100x10^{-5}}{1{,}000 * 1.8x10^{-5}} = 5v_{fs} \qquad (4)$$

Flow patterns depend on the room geometry and positions of heat sources and openings but not on the strengths of the sources. Temperature differences and flow rates do depend on source strengths.

To ensure similarity of convection patterns, geometric similarity should be observed and the Rayleigh number (which equals Gr Pr) should be between $2 \times 10^7$ and $2 \times 10^{13}$ [7].

In the auditorium occupancy heat gains (15 kW) were represented by 3 sources running the full length of the room. Since "summer" conditions were being modelled, no heat gains would occur from emitters provided for heating the space. Low level openings had a maximum area equivalent to 5.2 $m^2$ while chimney vents had an equivalent area of 2.6 $m^2$ leading vertically from the lecture theatre to a height of 19 m above the lecture room floor.

Displacement flow was observed (i.e. air entered at low level and exited via high level openings). Flow rates varied from 1.2 $m^3/s$ (with low and high level opening areas of 1.3 $m^2$ and a temperature difference of 10.8 °C) to 2.2 $m^3/s$ (with a low level opening area of 5.2 $m^2$ and a high level opening area of 2.6 $m^2$ and a temperature difference of 5.8 °C). The interface height varied from 3.0 to 4.5 m (this was the height of the high level "warm" zone above floor level).

It was found that adequate ventilation rates (i.e. that sufficient to provide sufficient cooling in "summer") could be achieved in most areas, including the auditorium. It was not advisable to use the larger concourse chimney vents to ventilate both the concourse and the theatre. The direction of flow between these two spaces was unpredictable and depended upon the actual heat gains and openings in each area. Thus there could be flow from the theatre to the concourse and vice versa.

An average dry resultant temperature could be calculated, by spatially averaging air temperatures and assuming the same surface temperatures (as were used in the ESP model). This was estimated to be 26.9 °C and the air change rate was 9.2 $ach^{-1}$. The corresponding values predicted by ESP were 27.6 °C and 6.6 $ach^{-1}$. This is deemed by the modellers to be a reasonable level of agreement.

## 1.9 Initial design calculations for auditorium 1.10.

The following equation [18] was used by the design engineers (Max Fordham & Partners) to calculate initial estimates of flow rates through auditorium 1.10 for particular average internal and external air temperatures to check to see if the fresh air requirements of occupants could be met by the stack effect:

$$q_v = C_d A \sqrt{\frac{2\Delta P}{\rho}} \qquad (5)$$

where $q_v$ = flow rate ($m^3/s$)

$C_d$ = discharge co-efficient (0.61c for sharp edged orifice)

$\Delta P$ = Pressure difference across opening (Pa)

$\rho$ = density of air (1.2 $kg/m^3$ at 20 °C and one atmosphere)

The space itself can be crudely represented as follows:

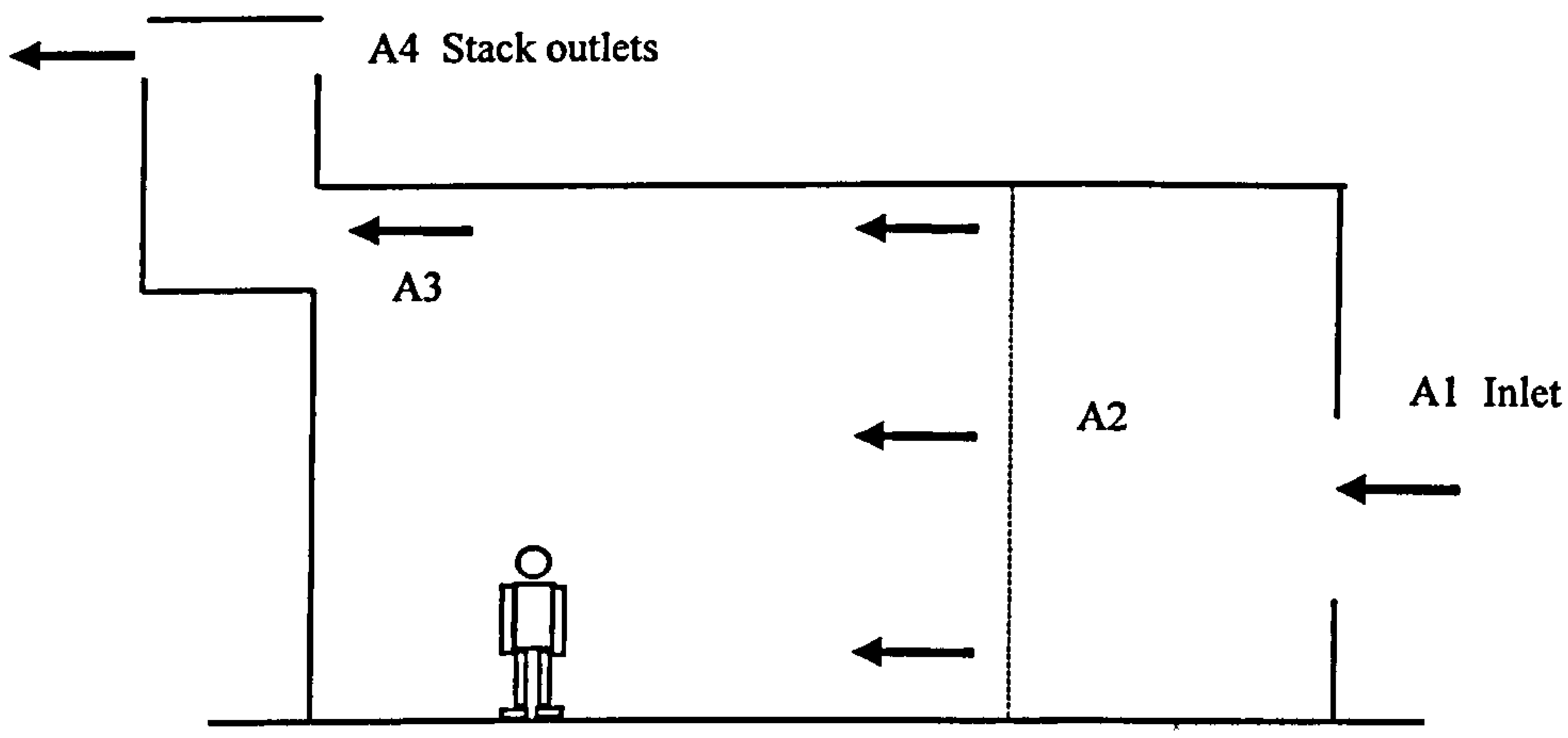
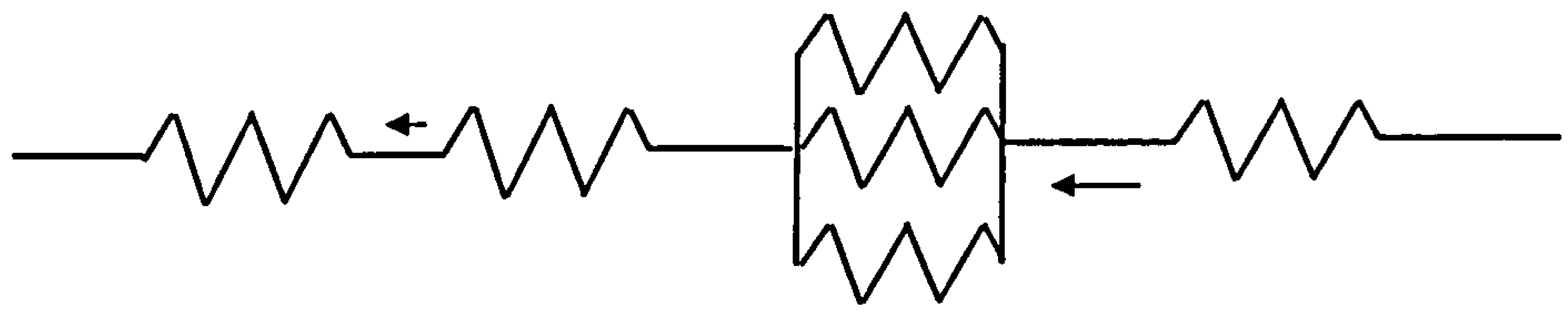



Figure 1.5 Section through a model of the theatre

Flow through the auditorium can be represented by flow obstructions connected in series and parallel:




Required stack pressure drop = $\Delta P = \Delta P_i + \Delta P_{ig} + \Delta P_{si} + \Delta P_{so}$ (6)

where $\Delta P_i$ = pressure drop across inlets

$\Delta P_{ig}$ = pressure drop across inlet grilles

$\Delta P_{si}$ = pressure drop across stack inlets

$\Delta P_{so}$ = pressure drop across stack outlets

Equivalent area of openings is calculated from [18]:

$$\frac{1}{A_E^2} = \frac{1}{A_i^2} + \frac{1}{A_{ig}^2} + \frac{1}{A_{si}^2} + \frac{1}{A_{so}^2} \qquad (7)$$

Thus, if initially areas of 3 $m^2$, 10 $m^2$, 3 $m^2$ and 3 $m^2$ are used for the inlet (external), grilles, stack inlets and outlets respectively, the effective area equals 1.7 $m^2$. Assuming internal and external air temperatures of 27 and 25 °C respectively (27 °C can be taken as the highest "allowable" room air temperature in a naturally ventilated building, and the outside temperature is a 2 (or 3) degrees less than the internal temperature so that air flows from the low level to the high level vents; this can be takén to be summer "worst case" conditions; i.e. a minimum temperature difference and no wind effects), and a stack height of 13 metres, then the stack pressure is found from the following:

$$\Delta P = (\rho_c - \rho_h)g(h_2 - h_1) \qquad (8)$$

$$\rho_c = \frac{P_a}{RT_{ao}} \quad ; \qquad \rho_h = \frac{P_a}{RT_{ai}}$$

$\Delta P$ = stack pressure drop (Pa)

$\rho_c$ = density of outside air (kg $m^{-3}$)

$\rho_h$ = density of room air (kg $m^{-3}$)

$h_1$ = height of inlets above datum (m)

$h_2$ = height of outlets above datum (m)

$P_a$ = atmospheric pressure; average value is 101.325 kPa

R = gas constant for air = 287 J $kg^{-1}$ $K^{-1}$ .

$T_{ao}$ = temperature of outside air (K)

$T_{ai}$ = temperature of room air (K)

Thus $$\Delta P = \frac{P_a}{R}\left[\frac{1}{T_{ao}} - \frac{1}{T_{ai}}\right]g(h_2 - h_1) \qquad (9)$$

$$\frac{P_a}{R}g = \frac{101{,}325}{287}9.81 = 3{,}463\,;$$

$(h_2-h_1) = h_a$.

$$\Delta P = 3463h_a\left(\frac{1}{T_{ao}} - \frac{1}{T_{ai}}\right) \qquad (9)$$

Thus:

$$\Delta P = 3463 * 13 * \left(\frac{1}{298} - \frac{1}{300}\right) \qquad = 1.0\ \text{Pa}$$

$$\text{Flow rate} = q_v = C_d A \sqrt{\frac{2\Delta P}{\rho}}$$

$$= 0.61 * 1.7 * \sqrt{\frac{2*1.0}{1.2}}$$

$$= 1.34\ m^3/s$$

Required fresh air flow rate = number of occupants x 8 litres/sec

= 150 x 8

= 1.2 $m^3/s$

Thus the stack flow rate can provide the fresh air requirements of a fully occupied auditorium (using the above areas and conditions)

If the temperature difference drops to 1 °C, then the flow rate reduces to 0.95 $m^3/s$. Increasing the inlets , stack inlets and outlet areas to 5 $m^2$ gives a flow rate of 1.55 $m^3/s$.

| $T_i$ (°C) | $T_o$ (°C) | ΔT (°C) | $q_v$ ($m^3/s$) |
|---|---|---|---|
| 27 | 25 | 2 | 1.34 |
| 27 | 24 | 3 | 1.65 |
| 25 | 20 | 5 | 2.15 |
| 20 | 10 | 10 | 3.12 |

Table 1.1 Stack generated flow rates for various internal and external temperatures, assuming a stack height of 13 m and an effective opening area of 13 $m^2$.

## 1.10 Rationale for monitoring the environmental performance of the Queens Building

Previously constructed buildings have shown ([1], [2]) that the use of natural ventilation and thermal mass can be effective in producing thermally comfortable conditions in UK climatic conditions and their requirements for energy are significantly less that that needed for mechanically ventilated and air conditioned buildings. However the expected occupant heat gains were significantly greater than had been designed for in other naturally ventilated buildings so calculations and modelling needed to be carried out to check to see if the building ventilation system would work with heat gains at and above 100 W $m^{-2}$. As seen above, a sophisticated dynamic thermal model was run during the design phase to calculate fabric and internal temperatures, while a reduced scale model was used to examine flow fields and global ventilation rates. Every model, by definition, can only examine some of the parameters that affect internal environments, so the only real way to validate a particular design is to test the prototype itself.

In summary, the main reasons for the measurement program (as outlined in this thesis) were as follows:

- To demonstrate that adequate ventilation rates and thermally comfortable conditions could be produced within a naturally ventilated auditorium, in spite of the very high occupancy gains that would occur

- To provide measurement data that could be compared to results obtained using thermal models, reduced scale (physical) simulations and CFD (computational fluid dynamics) programs

- To allow simple ventilation and thermal models to be developed which could be used to predict ventilation rates and internal conditions in similar enclosures

- To allow the building operators to fine-tune the ventilation control system such that the number of hours in the year when the enclosure is thermally uncomfortable could be minimised and that the energy use of the system could be reduced

- To highlight the particular design and operational problems of naturally ventilated, thermally heavyweight auditoria which have high intermittent heat gains

## 1.11 Contribution to knowledge on the design of low energy auditorium ventilation systems

No measurements of internal conditions (i.e. air and surface temperatures, room air speeds, indoor air quality, and bulk flowrates) have been carried out before in large, naturally ventilated auditoria. The use of natural ventilation to supply fresh air and for cooling is very unusual in auditoria. Usually mechanical ventilation and air conditioning are required because of the high internal occupant gains (c.100 $W/m^2$) and the large variation in internal gains which can occur (between minimum and maximum occupancy levels).

The De Monfort auditorium is a unique enclosure and it produces a unique set of internal conditions. The modelling work completed indicated that a thermally comfortable environment and adequate air flowrates could be maintained within this type of space despite the very high internal gains that would take place.

It is important to be able to determine experimentally whether the auditorium does actually provide the above conditions and to derive empirical models from the data obtained which can then be used in the design of similar buildings in the future.

# 2 Experimental methodologies

## 2.1 Review of methods for measuring ventilation rates in buildings

### 2.1.1 Large enclosures

Experimental methods used had to be appropriate for the monitoring of environmental conditions in a "large enclosure". Large enclosures being examined by the IEA Annex 26 work "Energy Efficient Ventilation of Large Enclosures" ranged in size from 1,000 $m^3$ (the De Montfort Auditorium) to 100,000 $m^3$ (the Gjovik underground stadium). Annex 26 was an internal collaborative research project whose main objective was to develop methods for minimising ventilation energy use in large spaces while maintaining thermally comfortable conditions and adequate IAQ (indoor air quality) within those spaces. More information on this annex is provided in Appendix F.

A large enclosure can be defined as a space where:

- The occupied volume is much less than the total volume
- Significant vertical temperature gradients occur
- Thermal buoyancy has a dominant effect on air motion
- Downdraughts may be present
- The Rayleigh number (Ra = GrPr) is large

The primary concerns are conditions in the occupied zone and instrumentation is generally located there. The number of sensors required may be significant and the minimum number of a particular type of sensor are required to adequately carry out given sets of measurements.

The following measurements are required:

i) *Tracer gas injection and sampling using the decay method.*

This will allow local and room air-change rates to be determined over a set period.

ii) *Air speed monitoring.*

Speed sensors located in the stacks allow instantaneous flow rates to be determined; sensors placed within the room give a measure of room air speed

iii) *Direction of air movement.*

Paper strips suspended across stack openings indicate the overall direction of air movement. If ultrasonic sensors are used, both air speed and direction can be simultaneously determined.

iv) *Air temperatures (internal and external)*

A number of sensors need to be placed horizontally to record variations in temperature that may occur across the width of the space.

v) *Surface temperatures.*

Surface temperature readings will enable mean radiant temperatures to be estimated.

## 2.2 Experimental techniques for monitoring internal environments in buildings

The following pages will describe experimental techniques which are currently available for measuring parameters i) to v) above, plus pressure differences across obstructions, and for examining air flow fields within rooms. The descriptions will commence with the available methods then follow on with the techniques and procedures actually used in the Queens Building auditorium.

The following methods are used for measuring ventilation rates in buildings:

i) Tracer gas injection and sampling
   a) Decay method.
   b) Constant emission method.
   c) Constant concentration method

ii) Velocity measurement in ducts.

iii) Fan pressurisation method

To determine thermal comfort parameters, the following needs to be recorded:

iv) Local air speeds and directions;

v) Air and surface temperatures.

To find bulk flow directions/movements the following techniques can be availed of:

vi) Tracking air movement using balloons;

vii) Particle streak velocimetry.

## 2.3 Tracer gas injection and sampling

A tracer gas (a gas which is non-toxic and which can easily be detected by gas analysers) is injected into the space being tested. The basic flow equation (for a single zone) is as follows:

$$V\frac{dC}{dt} = Q(C_e - C(t)) + F \qquad (2.1)$$

where

$V$ = effective volume of space, $m^3$

$Q$ = air flow rate through space, $m^3/s$

(assumed constant over the duration of the experiment)

$C_e$ = external concentration of tracer gas

$C(t)$ = internal concentration of tracer gas at time t

$F$ = production rate of tracer gas by all sources within the space

The rate of change of gas concentration is measured with time, this can be used to measure the air change rate in the enclosure [8].

### 2.3.1 Decay method

Gas is injected into a space and mixed until a particular uniform concentration is achieved. No further gas is injected into the enclosure and the concentration is allowed to decay. The rate of decay gives a measure of the ventilation rate; a high rate of decay indicates that a high air-change rate is occurring.

Since the production rate is zero, and it is assumed that outside air contains no tracer gas, equation 2.1 reduces to:

$$V\frac{dC}{dt} = -QC(t) \qquad (2.2)$$

Thus

$$\frac{dC}{C(t)} = -\frac{Q}{V}dt \qquad (2.3)$$

$$\int_{C(0)}^{C(t)} \frac{dC}{C(t)} = -\frac{Q}{V}\int_{t=0}^{t=t1} dt \qquad (2.4)$$

where C(0) = concentration of tracer gas at time t = 0.

Thus

$$\ln C(t) - \ln C(0) = -\frac{Q}{V}t \qquad (2.5)$$

$$C(t) = C(0)e^{-\frac{Q}{V}t} \qquad (2.6)$$

$$-\frac{Q}{V} = N = \text{Number of air-changes per unit time}$$

This method can be automated; i.e. more tracer gas can be injected into a space once a certain concentration has been reached. Thus a series of tracer gas decay curves could be produced, allowing flow rates to be calculated over an extended period.

### 2.3.2 Constant emission method

Tracer gas is emitted at a constant rate into a space. The relevant equation is:

$$C(t) = \frac{F}{Q(t)} + [C(0) - \frac{F}{Q(t)}]e^{-\frac{Q}{V}t} \qquad (2.7)$$

If there is not tracer gas present before the start of the test, and if the air flow rate into and out of the enclosure is constant, then

$$C(t) = \frac{F}{Q}[1 - e^{-Nt}] \qquad (2.8)$$

where

Q = air flow rate through the enclosure ($m^3/s$)

$N = \frac{Q}{V}$ = air change rate per unit time

A given period of time is required for the tracer gas concentration to reach equilibrium. For a high air-change rate this period will be relatively short (e.g. if N equals 4, then the time required to reach 95 % of the equilibrium value will be 0.75 hours).

Once the equilibrium concentration has been reached, then the flow rate can be calculated from:

$$Q = \frac{F}{C(t)} \tag{2.9}$$

A disadvantage of this method is that relatively large volumes of tracer gas are required; this can make the method expensive if the detecting equipment isn't particularly sensitive.

The measurement of carbon dioxide concentrations in occupied rooms to find ventilation rates is a constant emission technique (assuming the occupants act as constant emitters of $CO_2$ (if their levels of activity and sizes are similar)).

### 2.3.2(a) The passive tracer gas method

The passive tracer gas method is also a constant emission method. Small emitters (e.g. perfluorocarbon tracer (pft) gas sources) are placed within zones of interest. Passive samplers collect the tracer gas from the air by diffusion. The average concentration over the test period is found from the quantity of tracer gas desorbed from the sampler and the above equation is used to estimate the average ventilation rate.

A pft source comprises a small metal tube about the size of a small pencil and open at one end. It contains a few cubic centimetres of liquid pft and is stopped with a silicone plug. The pft vapour diffuses through the plug at an approximately constant rate (which depends on the surrounding temperature).

The samplers are standard diffusion tubes containing a granular carbon-based powder and cost about £15-00 each (1993). Pft diffuses into the tube at a rate proportional to the concentration in the room air, and is held within the strongly absorbing bed.

This method is generally used for longer term measurement of ventilation rates (i.e. over several days or weeks)

Under conditions of variable ventilation rates and occupancy levels, the actual flow rates are higher than measured values. This negative bias can be overcome by using multiple injectors and samplers.

### 2.3.3 Constant concentration method

Here, a constant concentration is maintained in a particular control volume by periodically injecting tracer gas. The relevant equation (for one zone) is :

$$0 = Q(-C(t)) + F(t)$$

hence

$$Q = \frac{F(t)}{C} \qquad (2.10)$$

Thus if the injection flow rate is constant; then measuring the total injection time will allow the total injection volume to be calculated, and the ventilation flow rate can be readily estimated.

This method required the most sophisticated apparatus; i.e. a complicated control system employing feedback is required to maintain constant concentrations in the areas being analysed.

### 2.3.4 Errors associated with tracer gas measurement techniques

Errors in the measurement of flowrate using the above tracer gas techniques depend upon the following factors:

1) How well the tracer gas is mixed in the space. This can be checked by using a number of sampling points.
2) Inaccuracies in estimating gas concentrations (this depends on the particular detector used).
3) Whether flowrates are constant or variable. Errors will be lower if the flow rate is constant.
4) Sampling rate (if ventilation rates are high, a low sampling rate may produce very few points on a log(concentration) versus time graph. Hence the slope of a line using the least squares technique cannot be found very accurately.
5) Number and location of sampling points used. As many sampling points as possible should be used but there are obvious practical limitations on how many can be used.
6) Complexity of particular method. For example the constant concentration method is an automated technique where the supply of gas to a zone needs to be regulated; i.e. high quality valves are required and the accurate timing of the opening and closing of the valves is required.
7) Some tracer gases may be absorbed by certain materials found in buildings and this would lead to lower concentrations being measured in a space.
8) Inaccuracies in determining building/zone volumes.

A summary of likely errors is given in the following table:

| Method | Overall error (%) | Comments |
|---|---|---|
| Tracer gas decay | +/-10 % | |
| Constant concentration | +/- 5 %<br>+/- 10 % | Constant flow rate<br>Variable flowrate |
| Constant emission (PFT) | +/- 10 % | Constant flow rate<br>Constant occupancy levels (errors depend partly on source temperatures) |

Table 2.1 Errors in tracer gas measurement techniques [8]

## 2.4 Fan pressurization method

The fan pressurization method is used to measure infiltration (i.e. air leakage) across the fabric of a building. A fan is connected to a space via a removable door and pressurizes the space up to a pressure differential of 50 Pascals (between the space and outside). The flow rate across the envelope is measured; this gives a measure of the leakage rate independent of weather conditions.

Equipment needed include a fan, a pressure gauge, an orifice plate flow meter, and a smoke generator to detect leaks. The fan is temporarily connected to the building envelope and its flow rate is altered to produce a pressure drop of about 10 Pa across the building fabric. The flow rate from the building to the outside is measured using a calibrated orifice plate and manometer. The pressure is increased by intervals of 10 Pa, and further flow measurements are carried out until the pressure-flow characteristic of the building has been found (i.e. the values of k and n in the equation $Q = k(\Delta P)^n$, where Q equals flow rate, $\Delta P$ equals pressure drop). The flow direction is reversed to depressurise the building and the above process is repeated.

The number of air changes per hour can be calculated by dividing the average flow rate (pressurization and depressurization) by the building volume. This value can then be compared with values from other buildings to check its performance.

This method is generally applied to a whole building and is not used for estimating interzone air flows etc.

If the flow rate at 50 Pa has been estimated, the approximate infiltration rate can be calculated by dividing the 50 Pa flow rate by 20.

The number of air-changes per hour in actual buildings (average values) varies from about 3 (Swiss buildings) to 14 (UK buildings) at 50 Pa pressure difference.

Errors in this method are likely to be about +/- 5 % when measuring flowrates.

## 2.5 Measurement of instantaneous air flowrates and air speeds

### 2.5.1 Hot wire anemometers

Flow rates can be monitored using omni-directional hot wire anemometers (i.e. high accuracy is achieved for a range of air flow directions) if a suitable duct or ducts is/are available and if it can be assumed that all the air entering a space passes through the duct(s). Low air speeds (i.e. values below 0.3 m/s) necessitate the use of hot wire sensors rather than a pitot tube plus differential sensors.

The hot wire sensor acts as one arm of a Wheatstone bridge circuit. The sensor probe is kept at a temperature of 30 °C above ambient temperature and the power supplied to the probe depends on the air velocity only. Temperature is monitored using a NTC (negative temperature coefficient) thermistor.

Outputs are mean air speed, standard deviation of air speed, plus turbulence intensity. The device will also measure air temperature. Accuracy is +/- 5 % down to an air speed of 0.05 m/s.

Hot wire sensors are connected to an analyser and data is logged via software on a PC.

Measurement of room velocities requires the use of some other device to indicate flow directions, i.e. a smoke puffer used adjacent to a sensor.

### 2.5.2 Ultra-sonic sensors

The speed of sound travelling between two or more probes in measured. If the speed of sound is known at a particular temperature and the time taken for a high frequency sound to travel a known distance from a transmitter to a receiver is measured, then the speed of air can be calculated. If the air is moving in the same direction as the sound signal, then the air speed will be distance travelled over time minus the speed of sound. If six probes in total are used then both the air speed and direction of the air can be computed.

An ultrasonic sensor is used to measure both air speed and direction (it outputs x, y, z velocities) but the large amount of data produced requires the use of a personal computer per sensor. The ultrasonic sensor measures the speed of sound between three sets of two probes; this speed gives a measure of air velocity between the probes. This type of sensor is an order of magnitude more expensive than the hot wire device and has approximately the same accuracy (when measuring air velocity magnitudes).

These devices have a very high accuracy (+/- 5 %) but the cost of a basic probe is high (approximately £ 2,000 in 1995).

## 2.6 Temperature measurement

A number of devices can be used to measure temperature. These include thermocouples (which use two dissimilar metals and require a reference ice-point temperature), thermistors (whose resistances reduce with increasing temperature), semi-conductor type devices and platinum resistance sensors (which have a very linear temperature resistance characteristic). Thermistors are the preferred device, as reasonable accuracies can be achieved (+/- 0.1 °C) and they are relatively inexpensive. However their outputs are linear over a narrow range of temperatures and they need to be regularly recalibrated.

Thermistor or semi-conductor type sensors are normally installed in buildings and are interfaced with the BMS (building management system) during the fit out stage. These are used to monitor conditions in the building and to control plant items. However their accuracy is not high (typically +/- 2 %) if they haven't been calibrated against a more accurate sensor and they give an overall indication only of internal conditions.

Surface temperatures are measured with the same devices, but there needs to be good thermal contact between the sensor and the measured surface. This can be achieved by having a sensor with a flat surface and by using a conductive paste between the sensor and the particular surface.

## 2.7 Pressure measurement

The usual method of measuring pressure differences in a fluid is to place an obstruction in the flow (i.e. a sharp edged orifice plate) and to measure the pressure drop across the plate using a manometer. This can then be used to determine flowrate.

It is desired to measure pressure differences across obstructions (e.g. inlet grilles, louvres, windows) in a naturally ventilated building. Hence the overall driving pressure will be no more than 5 to 10 Pascals and an inclined manometer will be unusable at these low pressure differences. A highly sensitive pressure sensor is needed with an accuracy of at least +/- 1 Pa.

Differential pressure sensors which measure from 0 to 5 Pa with a linear voltage output of 0 to 10 V are available but these are not sufficiently sensitive to measure the very low and rapidly fluctuating pressure differences found in naturally ventilated buildings.

## 2.8 Visualization methods

### 2.8.1 Tracking air movement using balloons

Balloons containing helium and weighed to achieve neutral buoyancy are used to track air movement in rooms. Their movement is recorded using video cameras at known positions (at least two cameras must be used). The images thus recorded are downloaded onto a computer and the locations of the balloons and corresponding times are recorded. This data can be used to estimate balloon speeds and directions and hence the nature of flow of air through a space can be determined. Under laboratory conditions, it is claimed that a positional accuracy of +/- 30 mm over a range of 8 m and the accuracy of 3-D velocity measurements of +/- 0.05 m/s can be achieved [9]. The experimental set-up is shown in figure 2.1 and the accuracy of the method is indicated in figure 2.2.

If a plane is photographed from several known positions, then the positions of objects within the plane can be determined from simple geometric and trigonmetric relationships. If the positions of the objects and corresponding times are recorded, the motions of the objects can be calculated. At least two views are required for positions to be found.

The motion of balloons are recorded using domestic quality video cameras and wide angle cameras placed at known, fixed positions. The images are then captured and analysed using a PC with a frame accurate playback deck and a video capture card.

The balloons used were made from metalised polypropylene, filled with helium. This material is fairly lightweight and the rate of diffusion of helium from such a balloon is low. The smallest feasible size was 22 cm, made from 15 μm thick polypropylene.

Some indication of the system output is shown in figure 2.3 (plan view of flow patterns in a gallery) and in figure 2.4 (projected sectional view of flow patterns in the gallery).

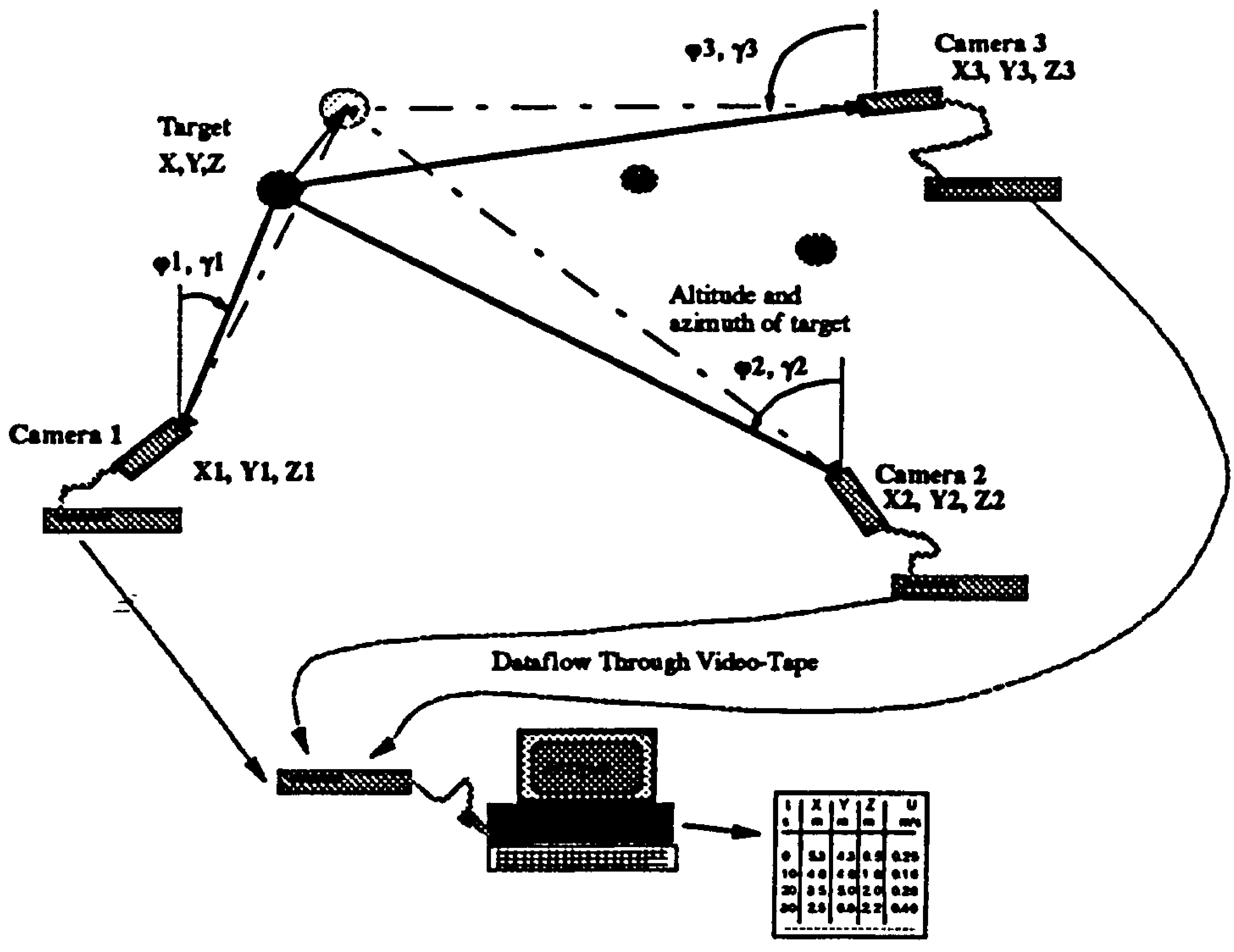



Figure 2.1 Functional block diagram of measurement and analysis system

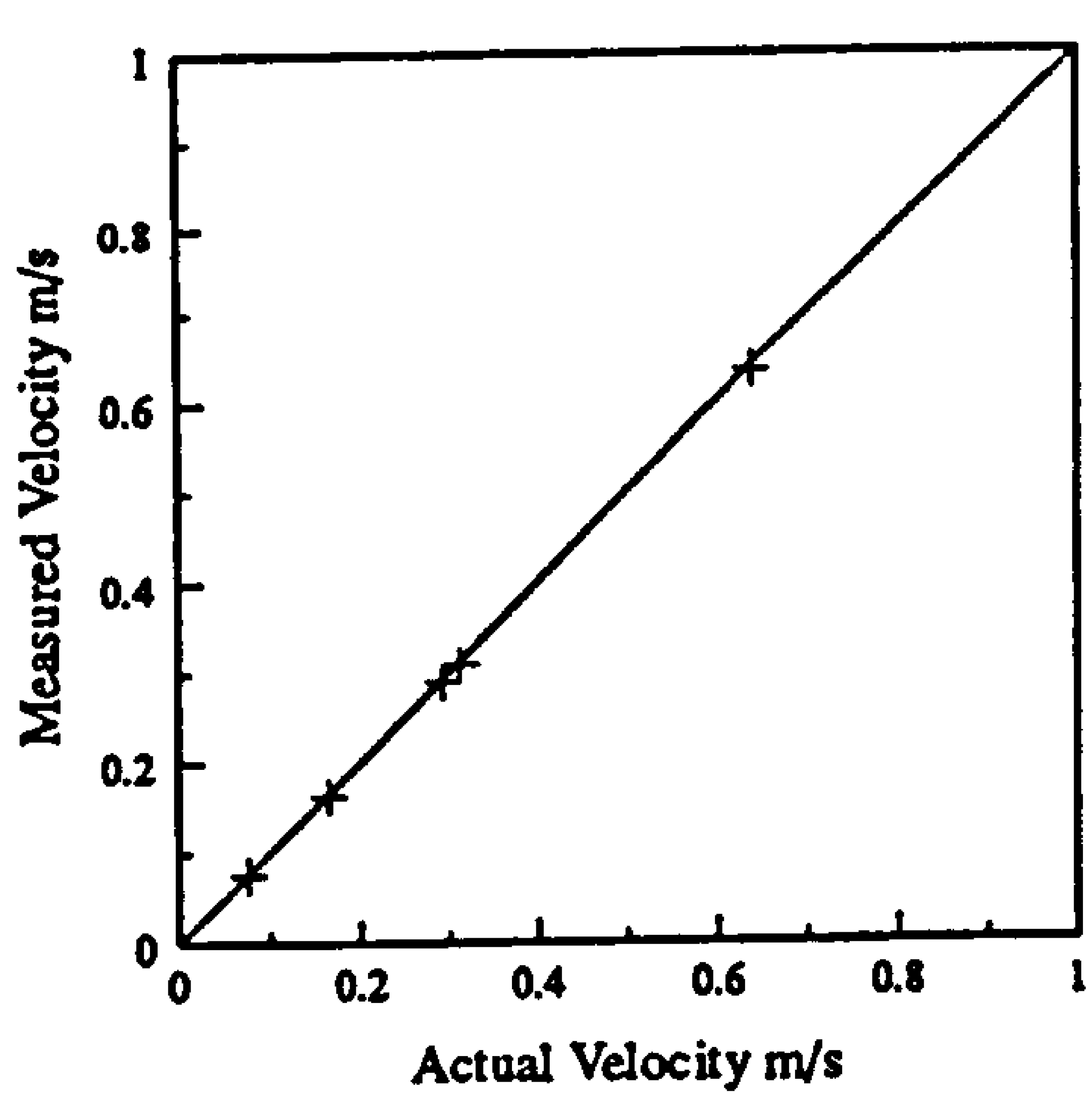



Figure 2.2. Comparison of measured and actual velocities

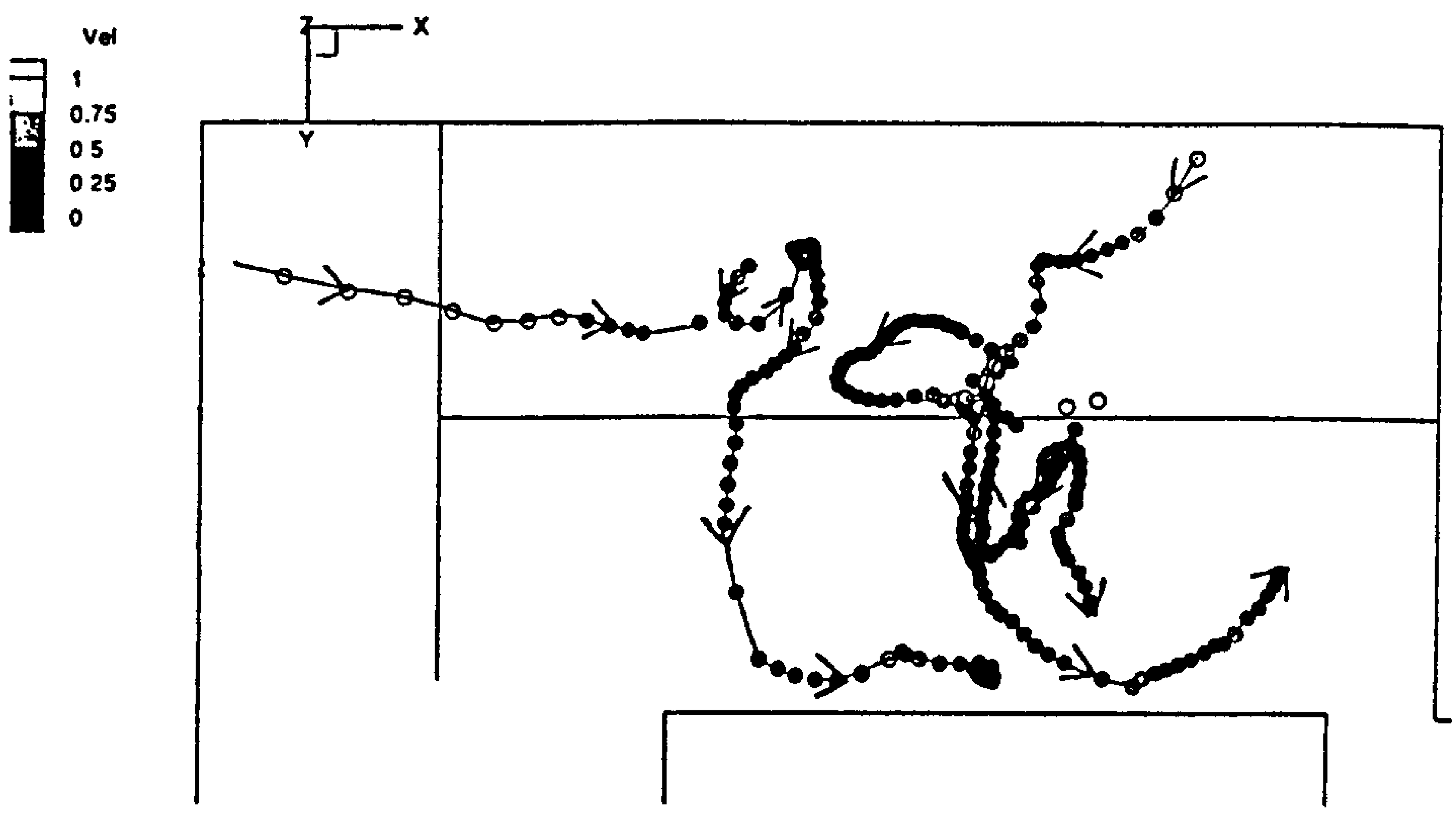



Figure 2.3. Plan view of flow patterns in a large gallery

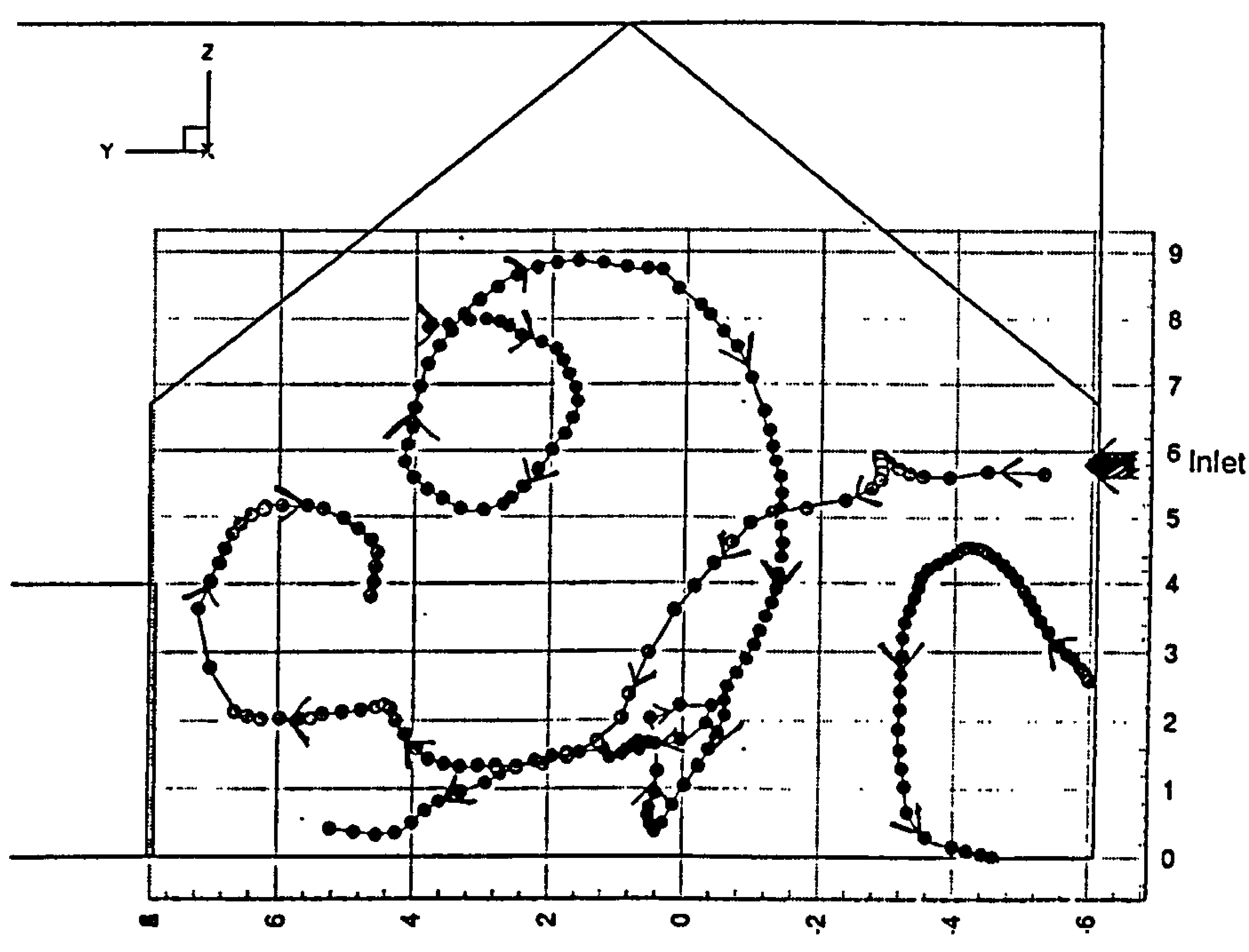



Figure 2.4. Projected sectional view of flow patterns in the gallery

#### 2.8.1(a) Errors

Errors can occur in converting the camera image to direction angles, converting to triangulation positions from these angles, in the timing of the images, and from the characteristics of the balloons themselves (deviations from neutral buoyancy etc.). Ideally the tracer should be physically small, have no inertial mass and maintain neutral buoyancy.

In practise it is very difficult and time consuming to achieve and maintain a satisfactory balloon weight.

Overall, the accuracy in velocity measurements is better than +/-0.05 m/s when using balloons moving under the influence of typical room air velocities (i.e. less than 1 m/s).

#### 2.8.1(b) Summary

The method is useful for measuring room air velocities in naturally ventilated enclosures with a reasonable degree of accuracy (+/- 0.05 m/s). Bulk air flow patterns/flow fields can be determined for a considerable period of time. The system is reasonably transportable and compares well in cost with air velocity measurements carried out with low velocity hot-wire anemometers. It compares favourably with the use of bubbles for visualizing flows; bubbles have very short life spans and leave a residue when they disintegrate. However a significant amount of time is required to set up positions of cameras and to correctly weigh balloons.

### 2.8.2 Particle streak velocimetry

This method involves using three cameras to record the movement of helium filled soap bubbles as they move under the influence of air currents within a space (see figure 2.5).

The bubbles are illuminated by a plane light sheet generated by a light source coupled with a special lens. The recording of images is done stereoscopically with three standard single lens reflex cameras by streak photography. The scanned negatives are analysed digitally [10].

Three dimensional instantaneous velocity fields can be produced.

Black and white film (ASA 3200) is used. To reproduce the three-dimensional co-ordinates out of two-dimensional images, at least two simultaneous recordings are required. However the information on flow directions cannot be obtained from only two views of a (non chopped) particle trajectory since the order in time is lost on the photographic film.

To get around this directional ambiguity, a third image is recorded with another camera. This additional camera is triggered simultaneously with the other two. The exposure time of the third camera is different, so that the streaks generated have only one of their end-points in common with the recordings of the other cameras. This common end point corresponds to the beginning of the recorded movement.

The scattered data can be interpolated to any point within the surveyed domain. A two dimensional interpolation function is used to independently interpolate the u, v, and w components of the velocity vectors. The interpolation can be improved if known boundary velocities are added to the measurement data (e.g. the zero velocities occurring at boundaries).

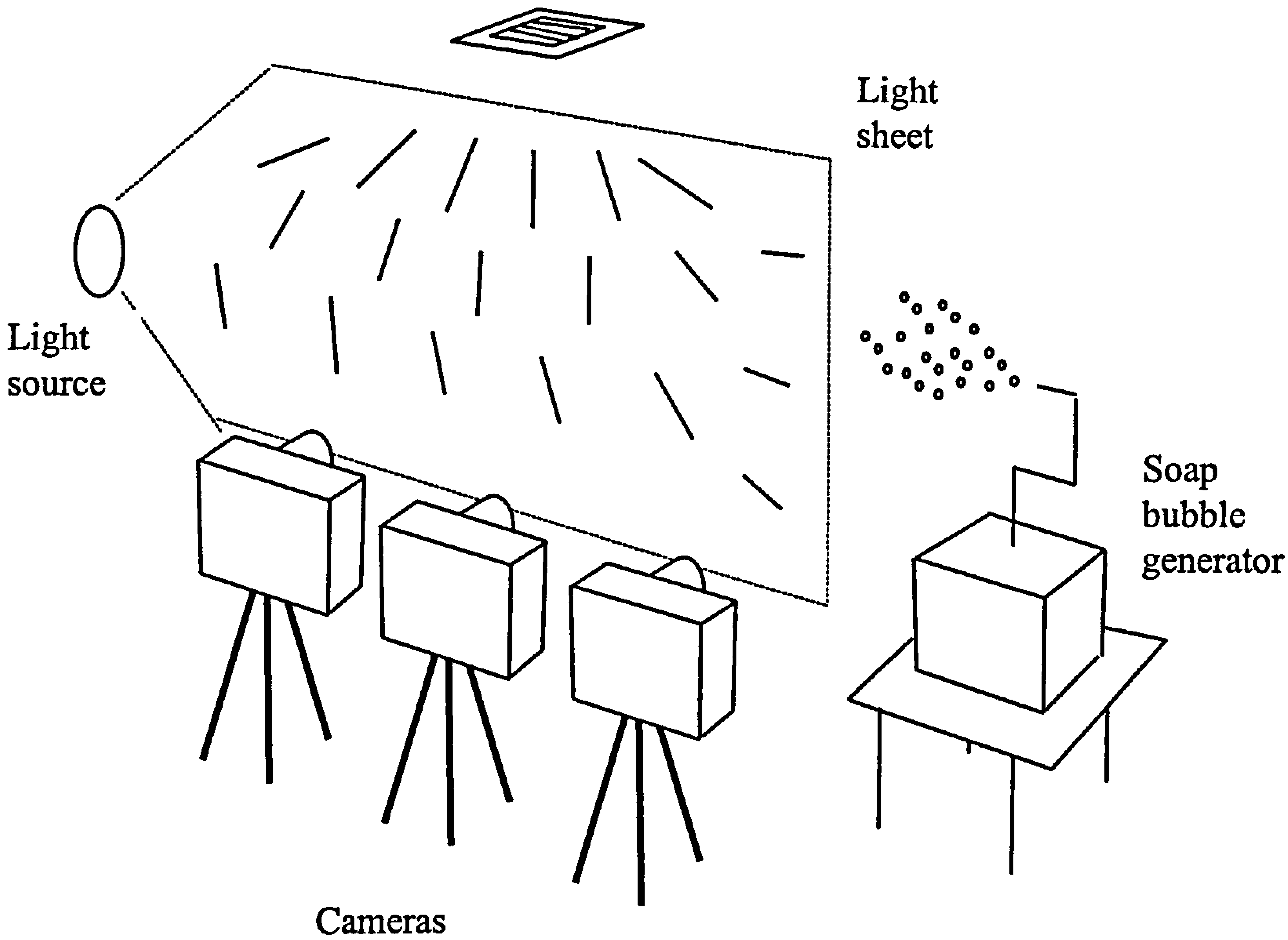



Figure 2.5 Particle streak velocimetry experimental set-up

## 2.9 Actual experimental methods chosen

### 2.9.1 Tracer gas methods

The decay or step down method was used to find local and average air-change rates. This allowed bulk time averaged flow rates and local air-change rates to be found. Measuring local air-change rates would enable ventilation efficiencies to be determined. Ventilation efficiency is a measure of how well a particular ventilation system distributes air within an enclosure from a separate source and how effectively it deals with the reduction of pollutant concentrations within the space. The method is relatively easy to implement, no sophisticated equipment is required, and less tracer gas is needed than required by the constant concentration method. In addition, equipment for implementing the method was readily available.

Initially, tracer gas was directly injected into the space for periods of up to 5 minutes or until the average gas concentration reached 50 ppm. Mixing was achieved using 6 oscillating desk mounted fans. After this period, samples were taken at 6 positions at one minute intervals and records were made of the gas concentrations. Recordings would be terminated when gas concentrations fell below c. 5 ppm. Further details of the use of the method are provided in section 3.4.

#### 2.9.1(a) Why the tracer gas decay method was used

A tracer gas method, in particular the decay method was chosen for measuring average flowrates. This method was used for the following reasons:

1) The method is suitable for measuring ventilation rates in naturally ventilated buildings. The instantaneous flowrate may vary considerably but an average value may be obtained using this method. Averaging periods varied from c. 0.5 to 3 hours.

2) The method was relatively easy to implement, no sophisticated equipment was required, and less tracer gas was needed than required by the constant concentration method (which is more suitable for smaller areas).

3) Equipment for implementing the method was readily available at no charge from the BRE; this organisation also provided training in the use of the decay method.

### 2.9.1(b) Tracer gas injection and sampling

The Autovent system was used to sample and log tracer gas concentrations. This was a constant concentration injection and sampling system developed by British Gas in the mid to late 1980's and consisted of a number of hardware units and control software written in BASIC and operated from a floppy within one of the original IBM PC's.

Hardware used comprised two injection units, a sample unit, an interface unit and an infra-red two channel gas analyser (which outputs a 0 to 1 Volt signal for gas concentrations from zero to 200 parts per million) (see figure 2.6 ). Injection of tracer gas was achieved by direct injection of gas (sulphur hexafluoride or $SF_6$) via mixing fans at points indicated in figure 2.6. Sampling tubes had been placed in both stacks, at seven points within the auditorium and in the inlet plenums. A thermistor was attached to the end of each room sampling line so that the air temperature of each zone could be recorded [11].

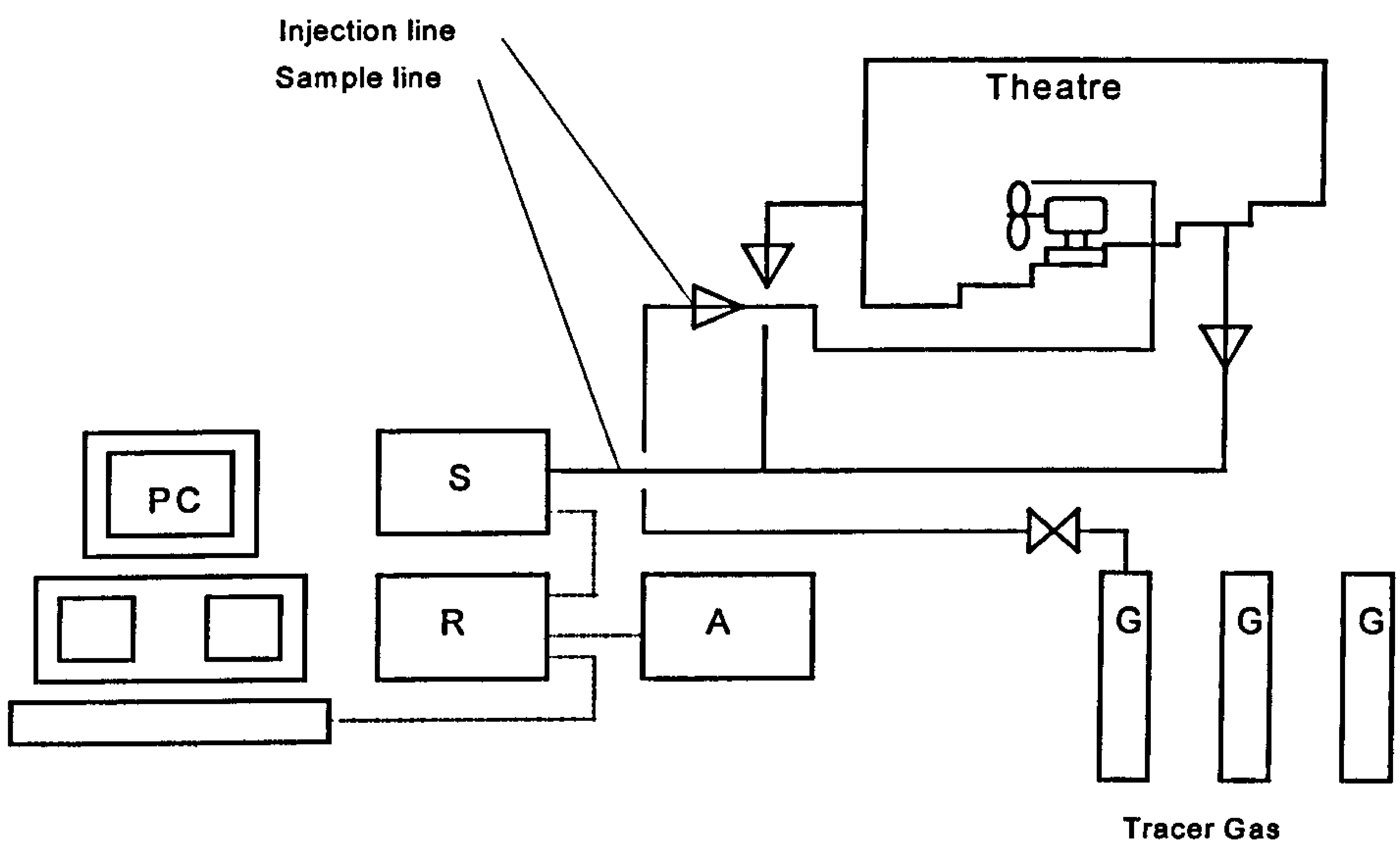



Figure 2.6 Schematic diagram of tracer gas injection and sampling system

S - sampling

A - analyser

R - Rexagan unit (control system hardware and interface unit)

PC - personal computer

Gas - $SF_6$, $SF_6$ in $N_2$, $N_2$

### 2.9.2 Air speed measurement

Air speed measurement was performed using Dantec low velocity anemometry instrumentation which allowed up to 24 velocities to be simultaneously recorded and logged [12].

An omni-directional probe is used; this acts as one arm of a Wheatstone Bridge circuit. The sphere is kept at a temperature 30 °C above ambient temperature and the power supplied to the probe is a function of velocity only. Temperature is measured using a NTC (negative temperature coefficient) thermistor which has a diameter of 0.5 mm and is suspended from a thin wire.

Outputs are mean air speed, standard deviation of air speed, turbulence intensity and air temperature. Measurement accuracy is +/- 5 % down to an air-speed of 0.05 m/s.

A PC connected via a parallel port was used as a controller and data logger. The system was very easy to operate via software supplied with the instrumentation.

24 15 m cables were supplied but the full number of sensors were not used because the long distances between sensors and the system necessitated using two cables for some positions. 4 of the sensors were mounted at a height of 9.5 m within each stack to allow bulk air flow rates to be measured. An additional 7 sensors were placed at low level in the occupied areas to measure air speeds and temperatures.

The system would only log the magnitudes of average air speeds and air turbulence thus some other means was required to monitor air direction. This was performed using a smoke puffer for room air speeds; paper strips suspended across stack openings were used to monitor the direction of air movement into and out of the stacks.

The instrumentation was very easy to set up and logging of data was straight-forward. The sensors were very sensitive and measured both air speeds and air temperatures, however they needed to be recalibrated annually.

Ultrasonic anemometers were also available, and two sensors were mounted at the stack inlet locations. Large amounts of data were generated and each sensor needed to be connected to a personal computer. These devices could measure velocities in the x, y and z directions and were used to record the directions of air flows into and out of the stacks.

A hand-held device was used to take spot readings of local air speeds.

### 2.9.2(a) Why the Dantec instrumentation and Ultrasonic sensors were used

1) The equipment was available at no cost and was provided from a relatively local source (Loughborough University).

2) Very low room air speeds were recorded. The only practical way of measuring these air speeds was to use a sensor based on the hot wire principle (such as the Dantec sensors).

3) The omni-directional probes allowed the simultaneous measurement of air speed and temperature.

4) The sensors and instrumentation allowed highly accurate and instantaneous values of air speed and velocity to be recorded.

5) The apparatus was very easy to set up and use.

### 2.9.3. Temperature measurement

It was possible to measure air temperatures using the Autovent equipment. A thermistor sensor was attached to each sampling tube (connected to the Autovent instrumentation) thus the Autovent equipment could log both local gas concentrations and zone temperatures.

A detailed series of measurements were performed to establish representative wall and ceiling positions for mounting surface temperature probes. These sensors (thermistors) were attached to four locations (two points on the ceiling slab and two wall positions) and data was logged at 15 minute intervals using a Squirrel logger. Two thermistor sensors were used to monitor the external temperature (logging period was 5 minutes). Sensors used were NTC thermistors with accuracies of +/-0.1 and +/-0.2 °C.

Temperature data was also available via the BMS computer (this could be down-loaded at weekly intervals).

### 2.9.3(a) Why the particular temperature measurement methods were chosen

1) Sensors used were required to be accurate (+/-0.2 °C for air temperature and +/-0.1 °C for surface temperature recordings respectively); and not be in need of regular recalibration (i.e. a recalibration interval equal to or greater than one year). The systems used also needed to be relatively inexpensive.

2) It was possible to connect up to 12 thermistor (temperature) probes up to the Autovent equipment and 12 sensors and cables were available. When the probes were installed, air temperature could be measured and recorded simultaneously with tracer gas concentrations.

3) Two Squirrel temperature loggers were provided by BRE so air and surface sensors that could be used with these loggers were purchased. The loggers were very easy to set up and use and the downloading operation produced ASCII files which could be readily imported into a spreadsheet program.

4) The BMS continuously logged temperature and other data. When this data was downloaded into a spreadsheet it was very easy to examine the thermal behaviour of the enclosure over periods lasting up to 6 days.

### 2.9.4 Pressure measurement

The pressure differential across the stacks was monitored using a Furness pressure sensor (0 to 5 Pascals) but due to the very low pressures recorded, it was difficult to obtain accurate readings. Hence very few pressure readings were actually obtained.

#### 2.9.4(a) Reasons for the use of the pressure sensing equipment obtained

1) The differential sensor was the most accurate sensor available at the time (accuracy of +/- 1 Pascal). It produced a linear output of 0 to 10 Volts for pressure differences of 0 to 5 Pascals.

2) The process logger used with the sensor was very easy to commission and operate.

### 2.9.5 Visualization of flow direction

Paper strips were suspended from cords positioned across the stack inlet openings. These gave a good visual indication of the direction of flow to/from the stacks.

A smoke puffer was used to find the direction of local air currents.

Balloons filled with helium and with neutral buoyancy were released in the seating area of the theatre. These rose vertically towards the ceiling and then moved across to the stack inlets. The vertical movement gave an indication that displacement flow was taking place.

### 2.9.5(a) Rationale for using the above equipment

1) The paper strips provided a simple but effective method for determining bulk flow directions. The main drawback was that an observer had to be present to record their movement.

2) The Dantec sensors were only able to measure air speed (not air velocity) so another method had to be used to detect the direction of air movement. A smoke puffer provided a simple, effective and low cost method of determining the direction of air flow at particular positions.

3) Balloons provided a means of detecting bulk air movement within a large enclosure. If the right sort of balloons are used with appropriate recording equipment, the method can be a very effective way of demonstrating bulk air movement.

### 2.9.6 Measurement of wind speeds and directions

Initially data on wind speed and direction was obtained via a weather station at East Midlands Airport. The anemometer was mounted in an exposed location at a height of 10 m. This data had then to be corrected for height (20 m) and location (city centre).

At a later stage, wind data was measured using a cup anemometer at a location on the roof of Gateway House, a tall building on the DMU city centre campus. The mounting height was approximately 26 m. The data was fed into a Squirrel logger. A sampling interval of 5 minutes was used, and data was downloaded once a month.

### 2.9.6(a) Why the above data source and equipment were used

1) A contact had been established at East Midlands Airport (EMA) who was happy to provide data on wind speeds and directions. The location was relatively close to Leicester.

2) It was found that the data provided by EMA was reliable and accurate when compared to data obtained from another site (Loughborough University).

3) BRE provided accurate probes for measuring wind speeds and directions plus an associated Squirrel logger. The sensors were installed on a mounting provided via the Estates Department of De Monfort University.

## 2.10 Overview of methods examined and used

Table 2.2 provides an overall summary of what methods were potentially available for the measurement of flowrate, infiltration, flow visualisation, air speed, temperature, and gas concentrations. Each method has been rated "good", "fair" or "poor" depending on its appropriateness when applied to the measurement program carried out in the Queens Building auditorium. An overall score has been assigned to each method, but giving a score of 3 to "good" methods, "2" to methods with a "fair" rating, and "1" to "poor" methods.

A score of "low" for costs has been assigned to the tracer gas decay method, velocity probes, and ultrasonic sensors as the equipment was provided free of charge by BRE or Loughborough University. Normally these systems have a high capital cost, particularly the tracer gas equipment (which requires the use of an infra red sensor, specialist sampling equipment, tubing and a PC). Velocity measurement in general is very expensive and the ultrasonic apparatus has a high cost, not only because the sensors are expensive (more than

£2,500 per sensor in 1995) but each sensor needs to be connected to a PC. This also applies to the omni-directional probe equipment.

| Measured parameter | Experimental method | Availability<br>Good<br>Fair<br>Poor | Costs<br>Low<br>Medium<br>High | Suitability<br>Good<br>Fair<br>Poor | Ease of use<br>Good<br>Fair<br>Poor | Accuracy<br>Good<br>Fair<br>Poor | Score (max = 15) | Method used (Yes/No) |
|---|---|---|---|---|---|---|---|---|
| Flowrate | Tracer gas | | | | | | | |
| | decay | √√√ | √√√ | √√√ | √ | √√ | 12 | Yes |
| | constant conc. | √√√ | √ | √ | √ | √√ | 8 | No |
| | constant emission. | √ | √√√ | √ | √ | √√ | 8 | No |
| | Velocity probes | √√√ | √√√ | √√ | √√√ | √√ | 13 | Yes |
| Infiltration | Tracer gas decay | √√√ | √√√ | √√√ | √ | √√ | 12 | Yes |
| | Fan pressurisation | √ | √ | √ | √√ | √√ | 7 | No |
| Flow visualisation | Balloon tracking | √ | √ | √√ | √ | √√ | 7 | No |
| | PSV | √ | √ | √√ | √ | √√ | 7 | No |
| | Smoke generator | √√√ | √√√ | √√√ | √√√ | √ | 13 | Yes |
| Air speed | Omni directional | √√√ | √√√ | √√ | √√√ | √√√ | 14 | Yes |
| | Ultrasonic sensors | √√√ | √√√ | √√ | √√√ | √√√ | 14 | Yes |
| | Heated thermistor | √√√ | √√√ | √√ | √√√ | √√ | 13 | Yes |
| (wind) | Cup anemometer | √√√ | √√√ | √√√ | √√√ | √√√ | 15 | Yes |
| Temperature | Thermistor probes | √√√ | √√√ | √√√ | √√√ | √√√ | 15 | Yes |
| | Thermocouples | √ | √ | √√√ | √√ | √√√ | 10 | No |
| | Resistance probes | √ | √ | √√√ | √√√ | √√√ | 11 | No |
| Pressure | Differential pressure sensor | √√√ | √√ | √ | √√√ | √ | 10 | Yes |
| Gas concs. | Infra red sensor | √√√ | √√√ | √√√ | √ | √√√ | 13 | Yes |

Table 2.2 Summary of methods available for evaluating internal conditions in the auditorium (good/low = √√√; fair/medium = √√; poor/high = √)

## 2.11 Summary and prelude to chapter 3

This chapter has described various methods for experimentally determining bulk flowrates, infiltration values, air speeds, surface and air temperatures, pressure differences and gas concentrations and techniques for visualising flows in rooms. It then details which methods were actually used in the auditorium and gives reasons why they were used.

Table 2.2 gives an overview of the above, and outlines a method for evaluating various experimental methods by assigning values to the appropriateness of each method (a "3" is given for "good" or "low" (cost), a "2" for "fair" or "medium" cost and a "1" for "poor" or "high" (cost). The methods achieving high scores can the techniques chosen.

Having specified which experimental techniques were used, chapter 3 will then describe the methodologies used with these techniques.

# 3 Experimental procedure

## 3.1 Objectives

The experimental objectives were to estimate the following for a wide range of external conditions:

1) Local and global (room) air change rates.
This would allow the effectiveness of the ventilation system to evaluated (i.e. how well is fresh air being distributed in the space for different sets of conditions). These are found from tracer gas measurements. Instantaneous flow rates are determined using stack mounted anemometers.

2) Environmental comfort related parameters.
Thermal comfort of a person at a location depends upon the following variables:

(i) Environmental variables:
- Air temperature
- Mean radiant temperature
- Horizontal and vertical surface temperature asymmetry
- Air speeds and direction
- Air turbulence

(ii) Occupant variables:
- Clothing levels ("CLO" value)
- Activity level (metabolic rate)

Mean radiant temperature can be found from mean surface temperatures or by recording the globe temperature and mean air speed. Dry resultant temperature is found from the average of the air and mean radiant temperatures (for air velocities below 0.1 m/s).

### 3.2 Methodology overview

The experimental methodology chosen had to achieve the above objectives for a large, naturally ventilated enclosure. Initially a list of the variables to be measured was required, this would then point to the equipment and systems needed to record these variables. It was important to determine what type of sensors and how many of a particular type would need to be obtained.

The enclosure could be divided into a number of zones; within the room there were low and mid-level (occupied) zones, and a high level (unoccupied) zone. Other zones included the two stacks, an underfloor plenum and three "fresh air " inlet plenums. It was deemed to be important to have at least one air temperature sensor in each of the room zones. A limited number of surface temperature sensors was available (i.e. 4); these were located on wall and ceiling surfaces adjacent to one of the rear exits and on the inner surface of the "external wall", and on a point on the ceiling a few metres from the wall. Checks (i.e. a series of measurements) were performed to ensure that the sensor locations used gave representative readings of average wall and ceiling surface temperatures.

The tracer gas sampling equipment allowed the simultaneous measurement of tracer gas concentrations and air temperatures, so a sampling point was also located in every "recorded" zone mentioned above. It was also possible to use this equipment to record wind speeds and directions; two connections were set up to take voltage inputs from specific devices but suitable probes and external mounts were not available.

It was considered important to measure flowrates through both stacks; if flow was relatively steady and uni-directional and if it was assumed that all extract air was removed via the chimney inlets, then stack air average velocities would give an accurate measure of global flowrates. An air speed probe had been positioned in the rhs stack a few metres from the outlets; however the readings from this sensor were virtually useless as it had been designed to operate air speeds from 2 to 10 m/s (speeds commonly found in mechanical supply and extract air handling plant). Inspection of the equation for stack induced flow will indicate a likely air speed range from 0.3 to 1.5 m/s; hence the probe would be operating in the lower end (and least accurate section) of its range. It is imperative when measuring natural ventilation flowrates that appropriate sensors are selected and used, i.e. those that have low errors in the selected range of air speeds likely to be encountered; have a very short response time (fractions of a second); are robust and experience little drift with time (to limit the effects of foreign bodies and minimise the need for recalibration). The air speed sensors actually used did fulfil the above requirements.

Under near ideal conditions (constant internal and external temperatures, no wind effects), steady flows could be produced. However, the no-wind condition is almost non-existent, so it was desirable to measure flowrates using two methods simultaneously (i.e. stack air velocity and tracer gas decay measurements.

It was important to measure both stack air speeds and flow directions; however the hot-wire type probes available would only measure average air speeds and air turbulence values. A method had to be found to detect (and possibly record) flow directions. One method would be the use of paper strips suspended from string stretched across the stack inlet openings, another would be the use of ultra-sonic sensors mounted on the opening ledges. The first method would necessitate the near constant visual inspection of the openings during an experimental run; the second would of course provide a permanent record of air velocities at points located in the centres of the openings. However the ultrasonic probes had to be operated via a PC and were available only for very short periods; so a very low technology method would have to be used to confirm air velocities. An additional method would involve the use of a hand-held smoke generator but this could only be used by an operator positioned at a height of 4m above floor level at a particular point in time (i.e. two persons would be needed to simultaneously detect air flow into both chimneys).

In practise, flow directions were monitored by viewing the suspended paper strips (see above) while an experimental run was in progress. A small number of runs were completed using the ultra-sonic probes mounted within the stack inlet openings.

It was considered important to be able to determine thermal comfort conditions in the space. Table 3.1 provides some data on conditions required for thermal comfort. A more rigorous evaluation of thermal comfort can be obtained by using equations provided in ISO 7730 [15]. This would require the measurement of air temperatures and velocities, and surface temperatures (or globe temperatures). It was not deemed to be important to continuously record internal relative humidity values; spot readings indicated that little variation occurred during occupied times. In addition, the allowable variation in relative humidity is quite large (CIBSE states a range from 40 to 70 % RH for offices). Surface or globe temperatures, coupled with air velocities would allow mean radiant temperatures to be calculated, and dry resultant temperatures would be found from the average of the air and mean radiant temperatures (for air speeds less than 0.1 m/s).

| Season | Dry Resultant Temperature range (°C) | Relative Humidity (range, %) | Air speed (max, m/s) | Horizontal temperature assymetry (max. temperature difference, °C) | Vertical temperature assymetry (max. temperature difference, °C ) |
|---|---|---|---|---|---|
| Winter | 20 to 24 | 40 to 70 | 0.15 | 10 | 5 |
| Summer | 23 to 26 | 40 to 70 | 0.25 | 10 | 5 |

Table 3.1 Approximate thermal comfort parameters for an office (assume a Clo value of 1.0 and an occupant is engaged in "light work", i.e. working at a desk in the office).

Vertical and horizontal temperature asymmetries could be determined from surface temperature readings and from spatially distributed air temperature sensors.

See appendix D2 for a list of apparatus used.

## 3.3 Experimental strategies

A large number of experiments (i.e. a least 4 "winter" runs; 11 "mid-season" runs and 8 "summer" runs) were carried out over a period of approximately two years. Tracer gas injection and sampling equipment together with tubing placed in 12 locations within the theatre (including sampling points in each stack) had been installed at the beginning of 1994. A number of preliminary tests were performed to check the overall operation of the Autovent system and to ensure that reliable data could be obtained. The Autovent injection and sampling apparatus was still in good working order (in spite of being out of use for c.

5 years); however the computer software and hardware was relatively primitive. Editing of the software was not possible and the complete system ran using one particular computer only.

The internal environment within the auditorium was to be monitored for a wide range of external conditions, i.e. for "winter", "mid-season" and "summer" conditions.

### 3.3.1 Strategy details

Details of strategies used for particular groups of experiments were as follows:

#### 3.3.1(a) Period: July 1994

A series of experiments were performed to measure room air speeds and stack air velocities over periods when high external temperatures would occur. The objectives were to examine the relative variation in air speeds at 7 locations within the space for varying inlet and outlet opening positions and to simultaneously monitor air flow rates.

### 3.3.1(b) Period: August 1994

Tracer gas testing (using the decay technique) was undertaken to allow local and global air-change rates to be calculated, when high outside temperatures occurred, for three user selectable opening settings (corresponding to low, intermediate and high effective opening areas).

On two consecutive days, smoke particles were released, and after complete mixing was achieved, particle numbers were counted to measure of the decay rates in particle concentrations. It was found that after c. 10 minutes, smoke was dispersed throughout the volume of the space (i.e. local air currents became undetectable). Air temperatures (external and internal) were also monitored.

Data on room, outside temperatures, wind speeds and directions was also being collected via the BMS. However, it was found that the wind speed and direction data was highly unreliable (when compared to data obtained at two other monitoring points within a 15 mile radius of Leicester), so it was decided to use data provided by East Midlands Airport only. These latter data sets were found to be reliable (when compared to figures obtained from Loughborough University) and personnel at the Airport were willing to provide this information.

### 3.3.1(c) Period: December 1994 to January 1995

Tracer gas concentrations; air temperatures, stack and room air speeds were recorded to find air distribution rates, to allow room comfort conditions to be estimated and to determine overall flow rates, for a range of inlet and outlet areas, when outside temperatures were low (less than 5 °C).

The phenomenon of stack down draughts would be monitored, when vents would move to their closed positions.

### 3.3.1(d) Period: January 1995 to April 1995

Local and global rates of air distribution, and temperatures were measured using previously described techniques, for moderate external air temperatures.

A series of runs were performed using simulated occupancy loads (50 100 W light bulbs plus mountings placed on seats). A later set of runs examined the ventilation performance using actual occupants (i.e. $CO_2$ levels were monitored). High ventilation rates could not be used when the space was occupied as high discomfort levels would result. The rate of decay of the $CO_2$ levels (post occupancy) was used to calculate infiltration rates.

### 3.3.1(e) Period: June to August 1995

A number of runs was carried out to establish local and average air change rates, room temperatures and air speeds, when high outside air temperatures were prevalent.

A large number of experiments was completed to examine the effects of wind direction and speed on ventilation rates. The stack outlets could be varied independently by manual control (using control boxes at the base of each chimney). These were set to particular configurations for approximately 15 to 20 minutes. Four different configurations were chosen (i.e. see figure 4.27).

Over a period of several hours, ventilation rates, room air speeds and temperatures were recorded. Hence the effect of varying the outlet "shapes" could be quantified (i.e. were ventilation rates increased when the open sections of the outlets were facing leeward?).

## 3.4 Experimental procedure

### 3.4.1 Use of the tracer gas decay method

Time averaged air change rates are found using the decay concentration method. Air change rates are calculated by integrating plots of gas concentrations versus time (hours) and dividing these values into the initial concentrations. Room air change rates were then found from the average of 6 local air change values. Actual air flow rates can then be readily found if the room volume is known.

After injection of the tracer gas, the operation of mixing fans ensured that the gas is well mixed throughout the space. Fans are run for about 20 to 25 minutes. Inlets and outlets are set in the desired positions (i.e. 50, 75 or 100% of the maximum settings) via the B.M.S., which correspond to effective areas of 2.43 , 3.18, & 3.57 m$^2$ respectively (see table 3.1 ). See figure 3.1 for the layout of injection and sampling points used.

| Opening | Width (m) | Height (m) | Free Area Fraction | Area (m$^2$) | Number | Total Area (m$^2$) |
|---|---|---|---|---|---|---|
| Inlets | 1.8 | 1.97 | 0.55 | 1.95 | 3 | 5.86 |
| Inlet grilles | 90.4 | 0.25 | 0.65 | 14.69 | | 14.69 |
| Stack openings | 1.56 | 1.96 | 1.0 | 3.055 | 2 | 6.11 |
| Outlets | 0.9 | 0.52 | 1.0 | 0.468 | 8x2 | 7.49 |

| BMS Setting (%) | Area (m$^2$) of Inlets | Inlet grilles | Stack openings | Outlets (stack top) | Total effective area (m$^2$) |
|---|---|---|---|---|---|
| 50 | 3.30 | 14.69 | 6.11 | 4.64 | 2.43 |
| 75 | 5.29 | 14.69 | 6.11 | 5.62 | 3.18 |
| 100 | 5.86 | 14.69 | 6.11 | 7.49 | 3.57 |

Table 3.2 (a) & (b) Opening areas

Note: BMS = Building Management System

Free area percentages :

Inlets; 80 %; Grille inlets; 65 %

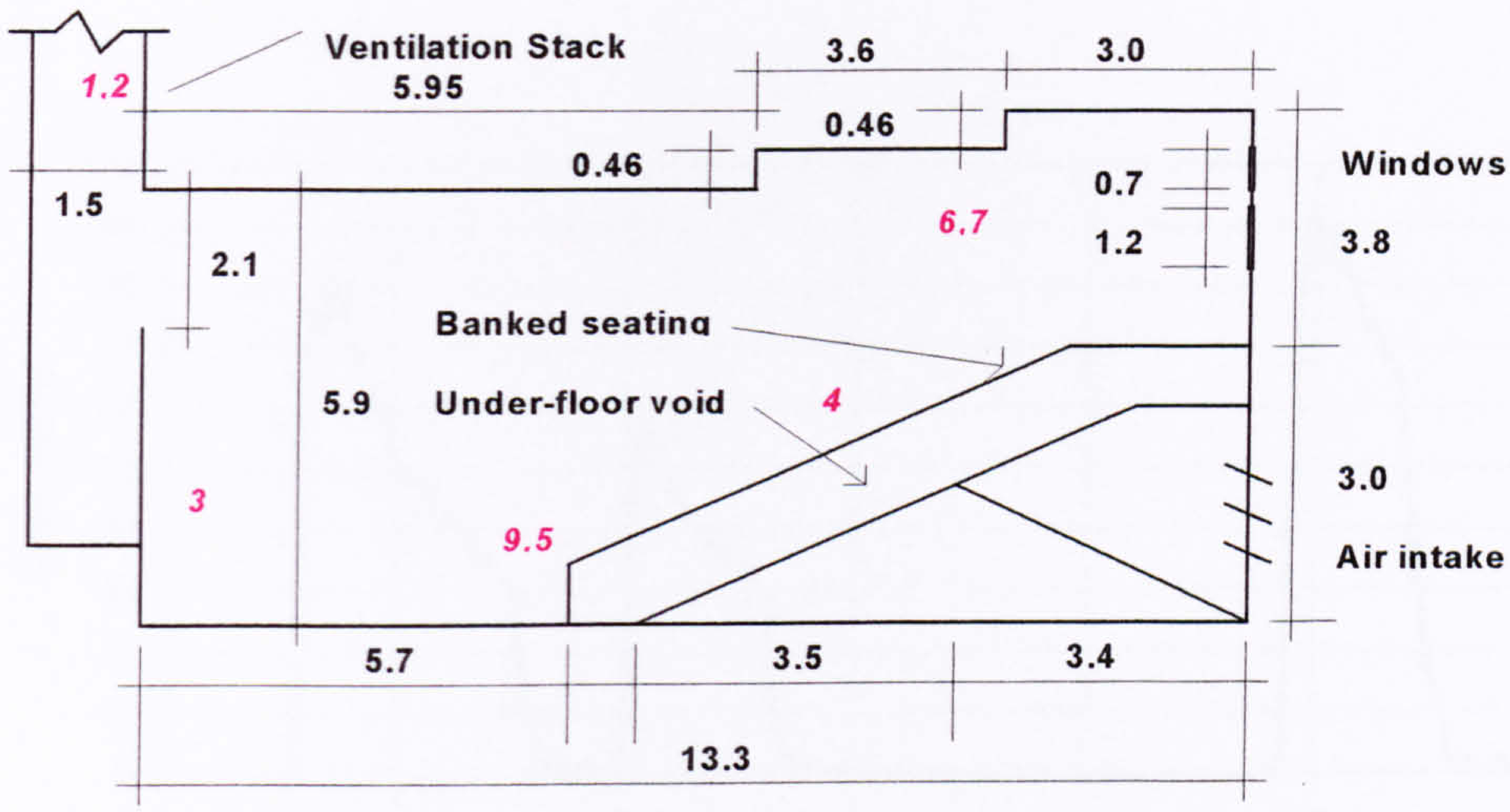
Stack opening; 100 %, Outlets (stacks); 100 %

Total effective area is for stack flows only (with inflows via low level openings and outflows via high level stack openings)

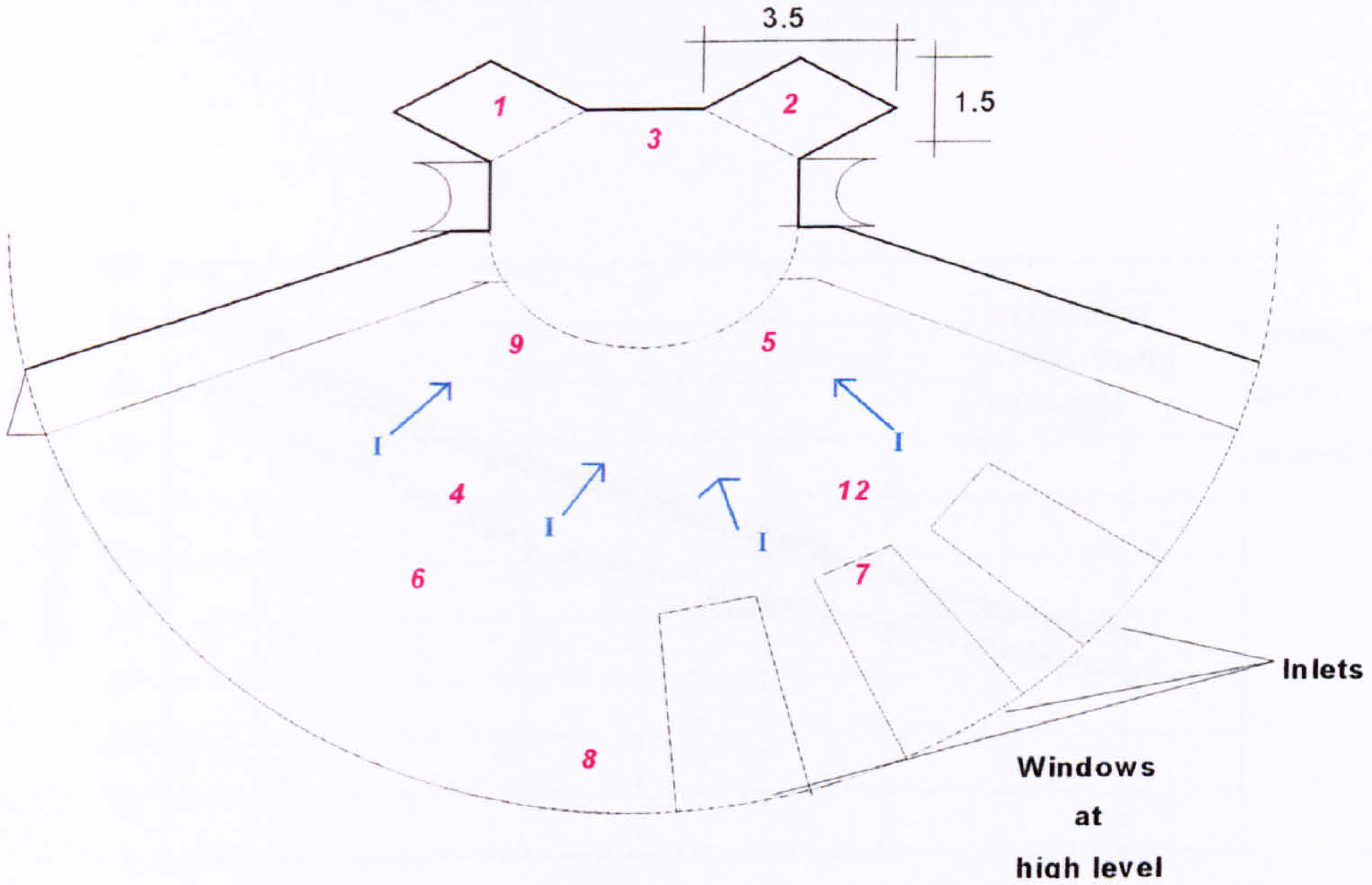
Figure 3.2(a) gives a plot of tracer gas concentrations versus time for a number of different opening settings. Figure 3.2(b) gives a more detailed view of figure 3.2(a) and demonstrates how average air-change rates were found (by fitting an exponental curve to the data; the exponent of the equation found is then the air-change rate). The average concentrations shown are less than the stack values as the average had been estimated from a total of 6 sets of data (6 measurement points).

Casual heat gains due to occupants of up to 5 kW were simulated using 100 Watt electric filament light bulbs; these loads were placed in the first 4 rows (nearest the lecturers position). This load represented an average occupancy value; the use of higher heat loads would have required additional power supplies.

See Appendix D1 for a detailed listing of the experimental methodology used.




**Section through theatre**




**Plan (all dimensions in metres)**

Figure 3.1 Section and plan of theatre 1.10 showing location of injection and sampling points. (Room area = 187 $m^2$, room volume = 870 $m^3$).

I correspond to injection points while underlined numbers shown give locations of sampling points.

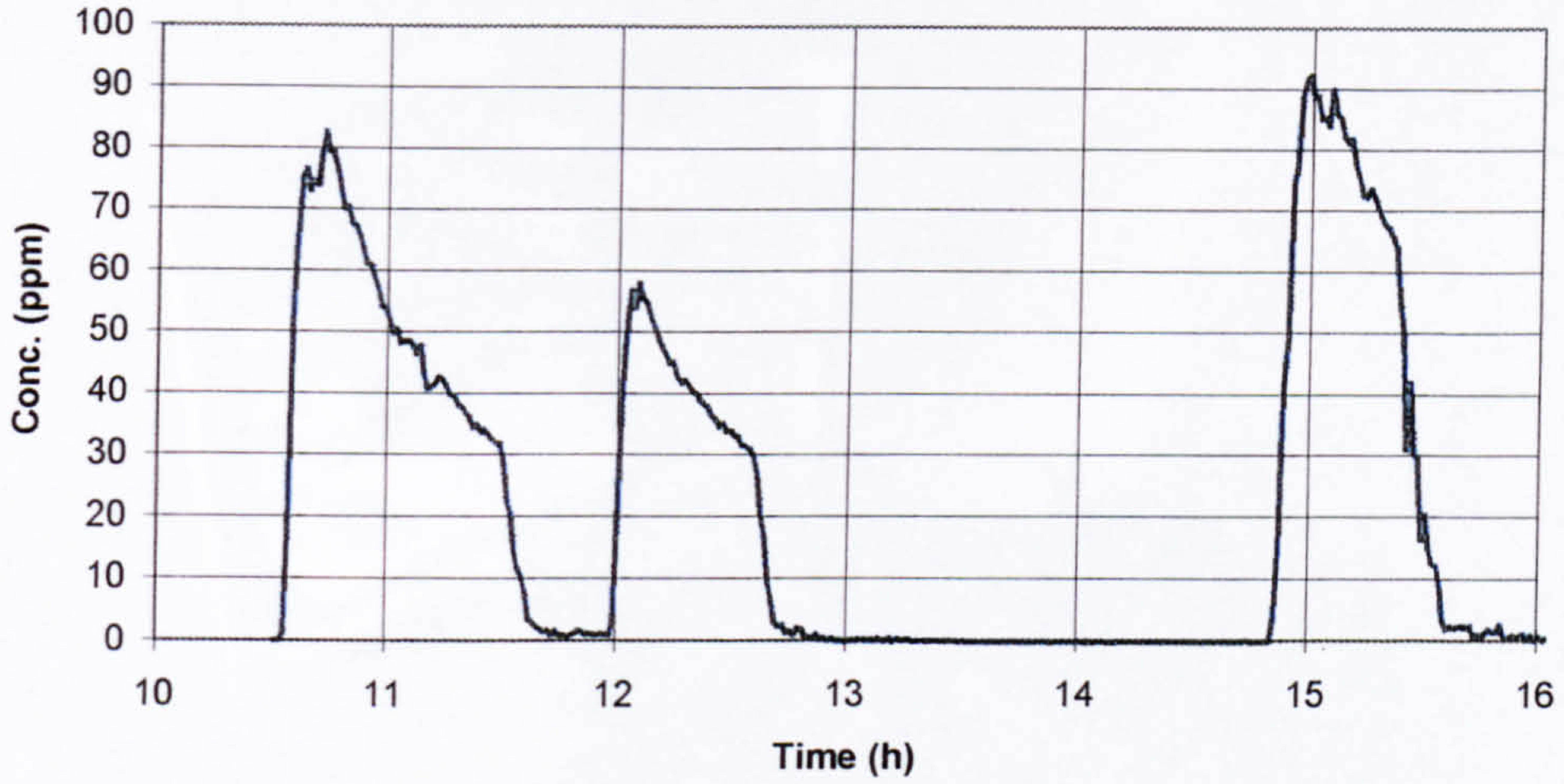



Figure 3.2 (a) Plot of tracer gas concentrations versus time in an auditorium (infiltration only; inlets and outlets closed)

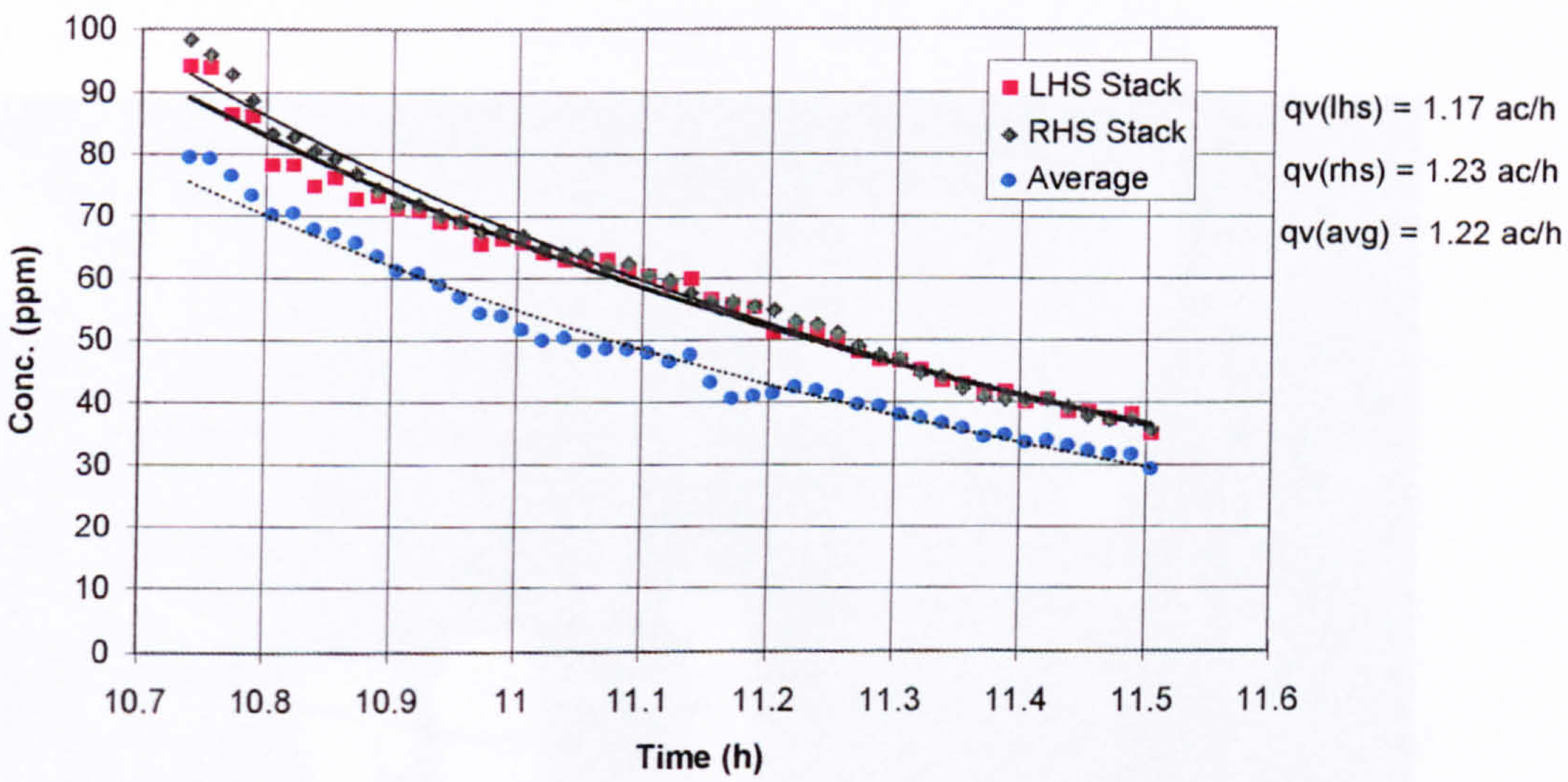



Figure 3.2 (b) Plot of tracer gas concentrations versus time; determination of air-change rates due to infiltration; 10.7 to 11.5 hours.

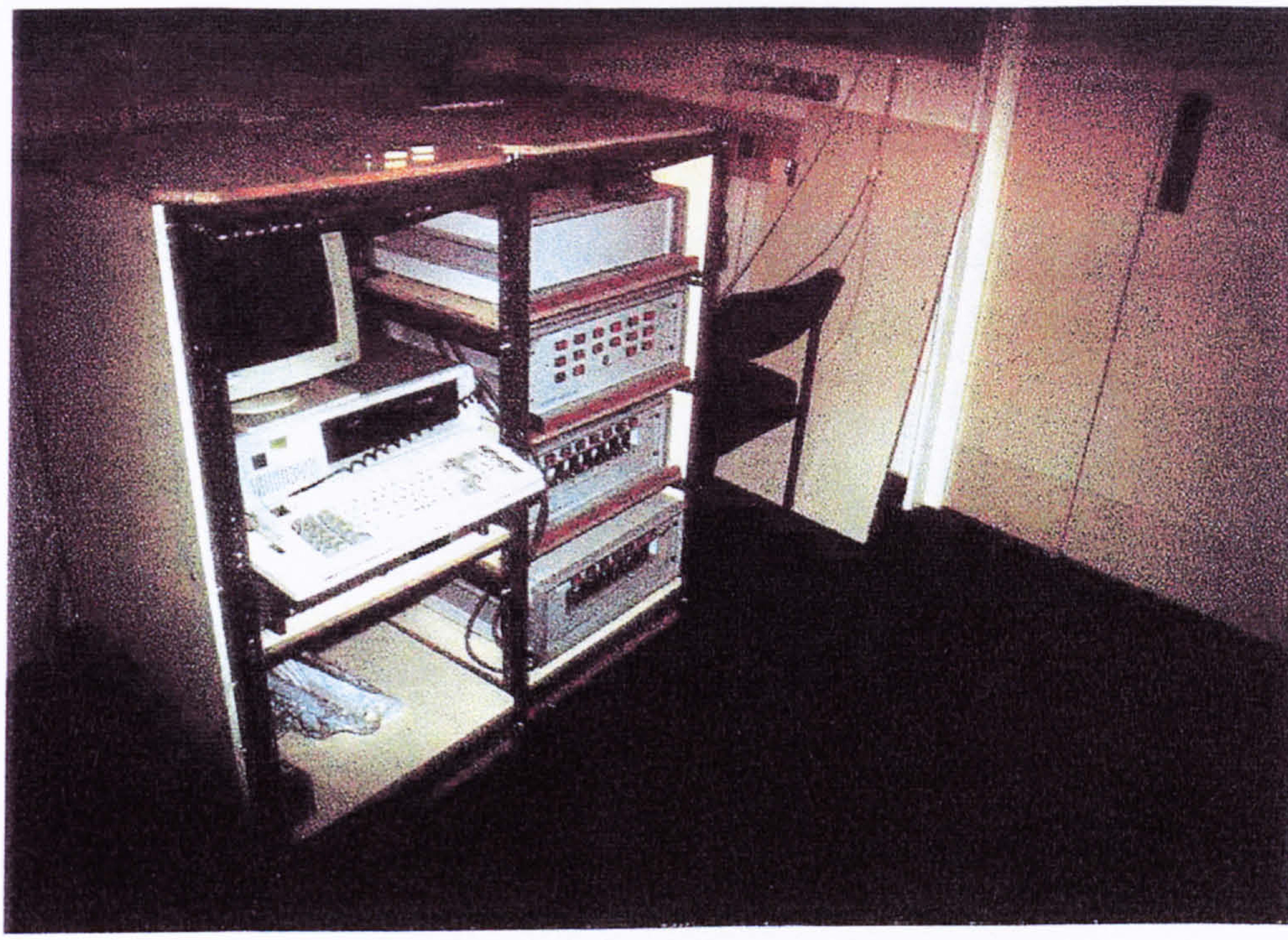


Figure 3.3 Autovent tracer gas sampling system

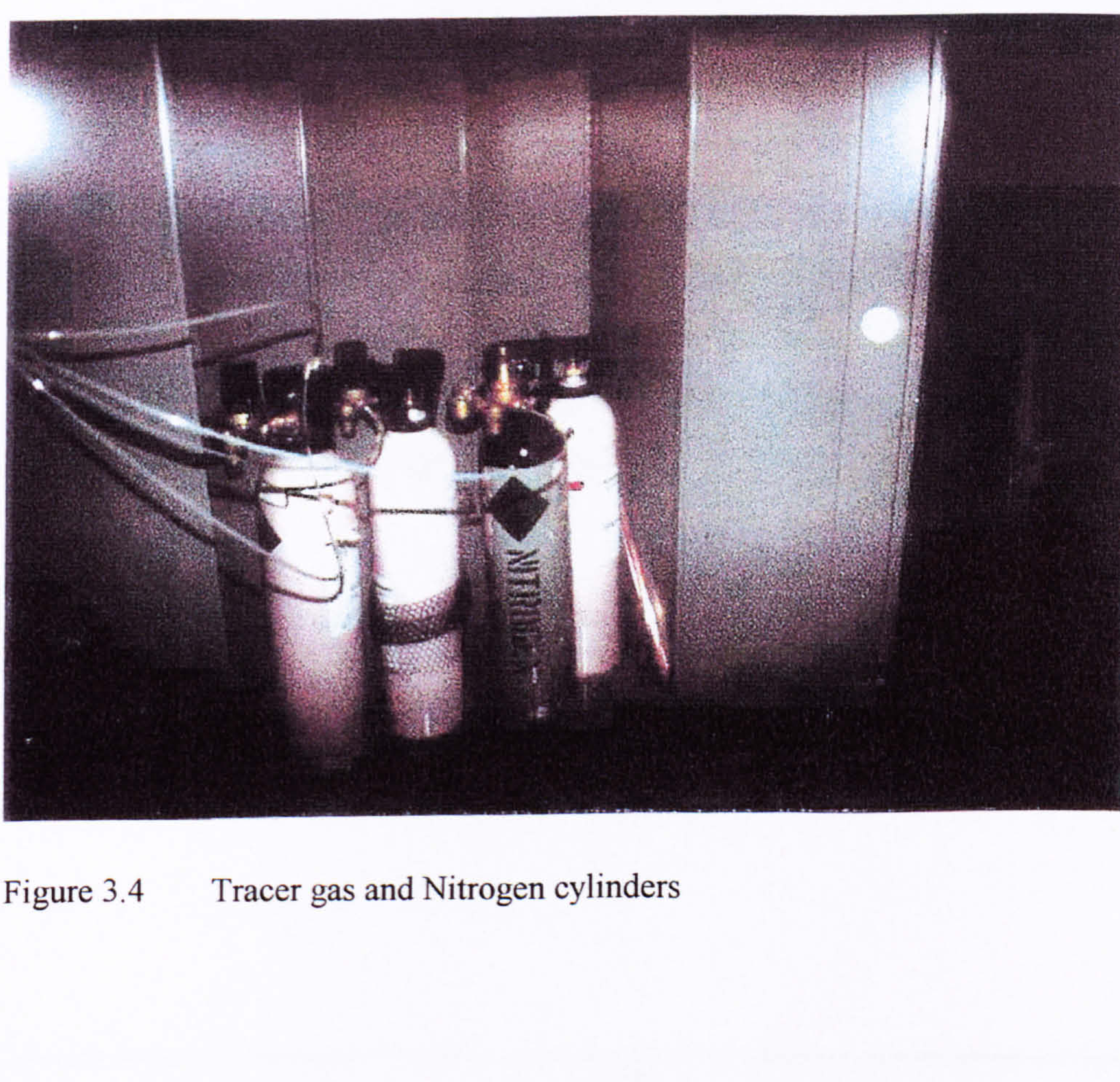


Figure 3.4 Tracer gas and Nitrogen cylinders

### 3.4.2 Velocity Measurement

Air velocities were measured at positions in the first and fifth rows 0.5 metres from inlet grilles (these were thought to be the most likely positions where discomfort would occur), and at four points within each stack, at heights of 9.25 m, using low velocity hot wire probes, in order to measure average flow rates (see figure 3.6). A structural beam was available at this height in the stacks across which a mounting for the velocity sensors could be supported. The turbulence existing at and near the entry openings to the stacks would have partially subsided at this height and thus more constant readings of air speeds could be obtained. Measurement periods were similar to those used for the tracer gas monitoring, in this case 16 to 75 minutes.

Flowrates in a duct are normally found by measuring velocities at an array of points (over the full width and depth of the duct at positions specified in BS 1042). It is usually assumed that velocities and hence flowrates are relatively constant when using this procedure. However measured velocities were highly variable; velocity profiles were not maintained over a measurement period (i.e. an interval of less than 3 hours) and it would have been almost impossible to carry out velocity traverses in practice.

Velocity directions were periodically checked using a hand-held smoke generator and/or by viewing the movements of strips of paper suspended across the inlet to each stack. Flow directions could also be determined from the probe temperatures; if the external temperature was considerably less than the room temperature, a sharp drop in the stack temperature would indicate that downflows were occurring.

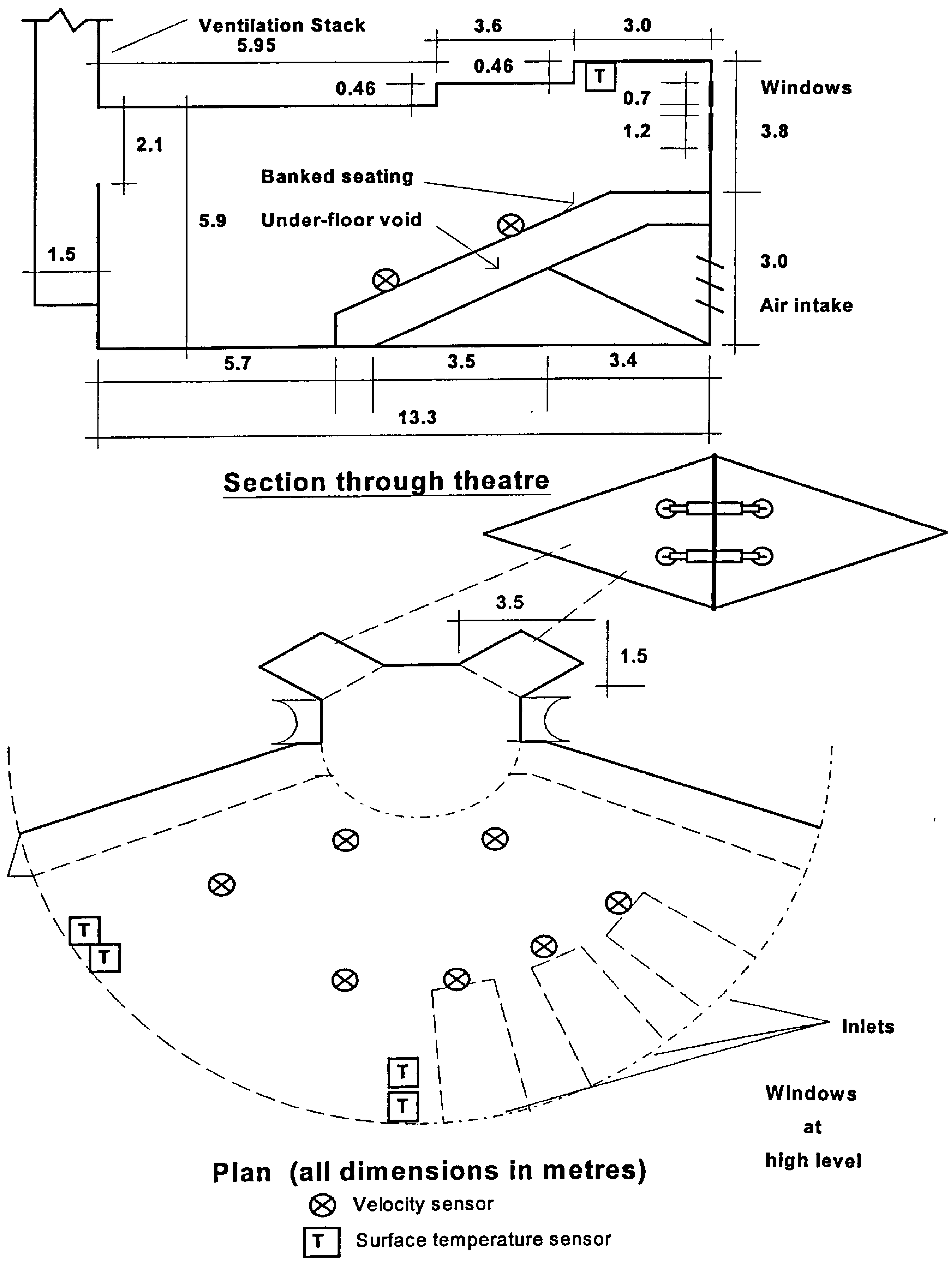



Figure 3.5 Section and plan of theatre 1.10 showing the positions of velocity sensors; wall and ceiling mounted surface temperature probes and section showing sensor mounting orientations in a stack

Figure 3.6 Dantec low velocity recorder plus PC controller and logger

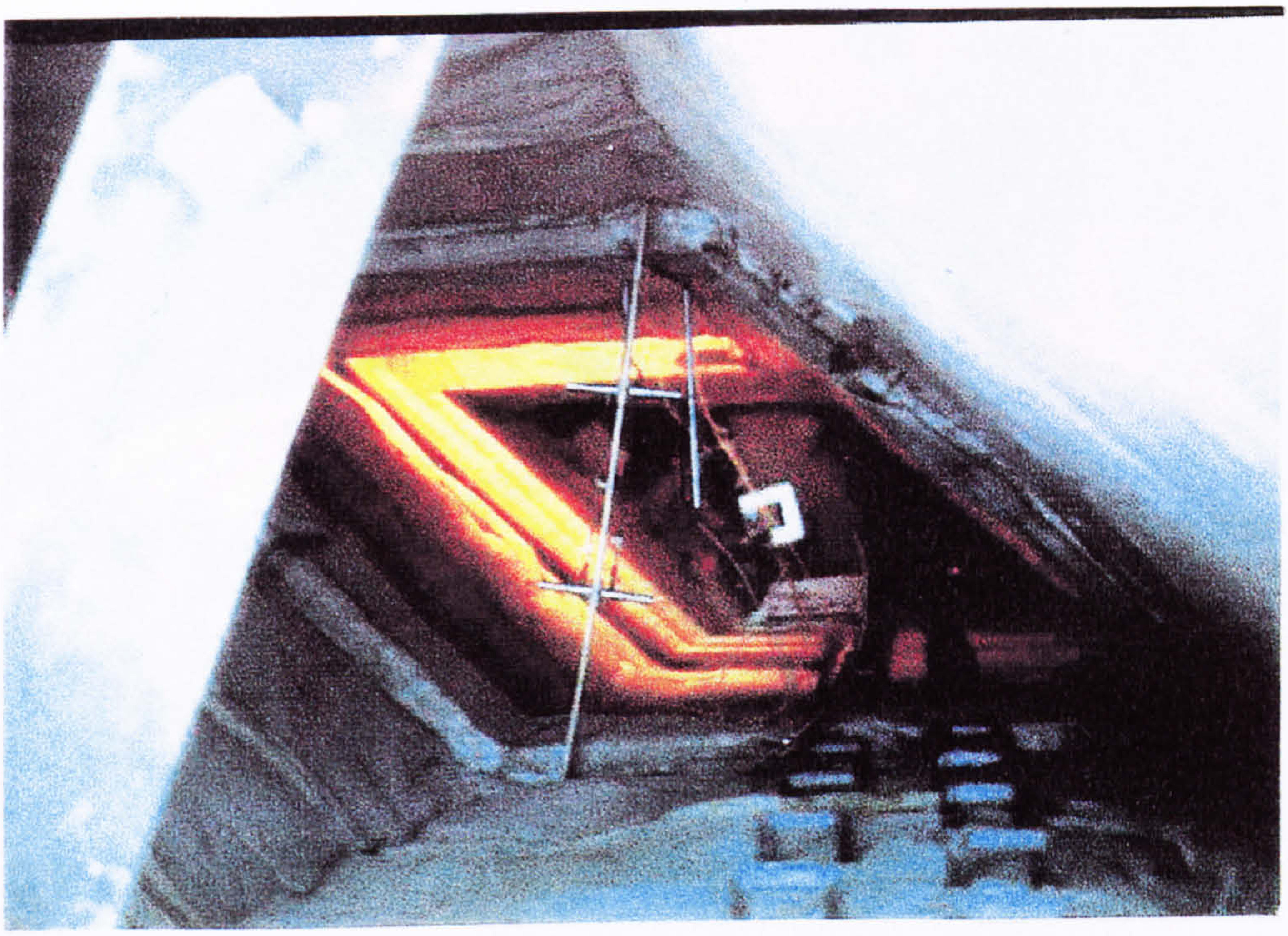


Figure 3.7 Low velocity sensors mounted at a height of 9.5 m in one of the stacks

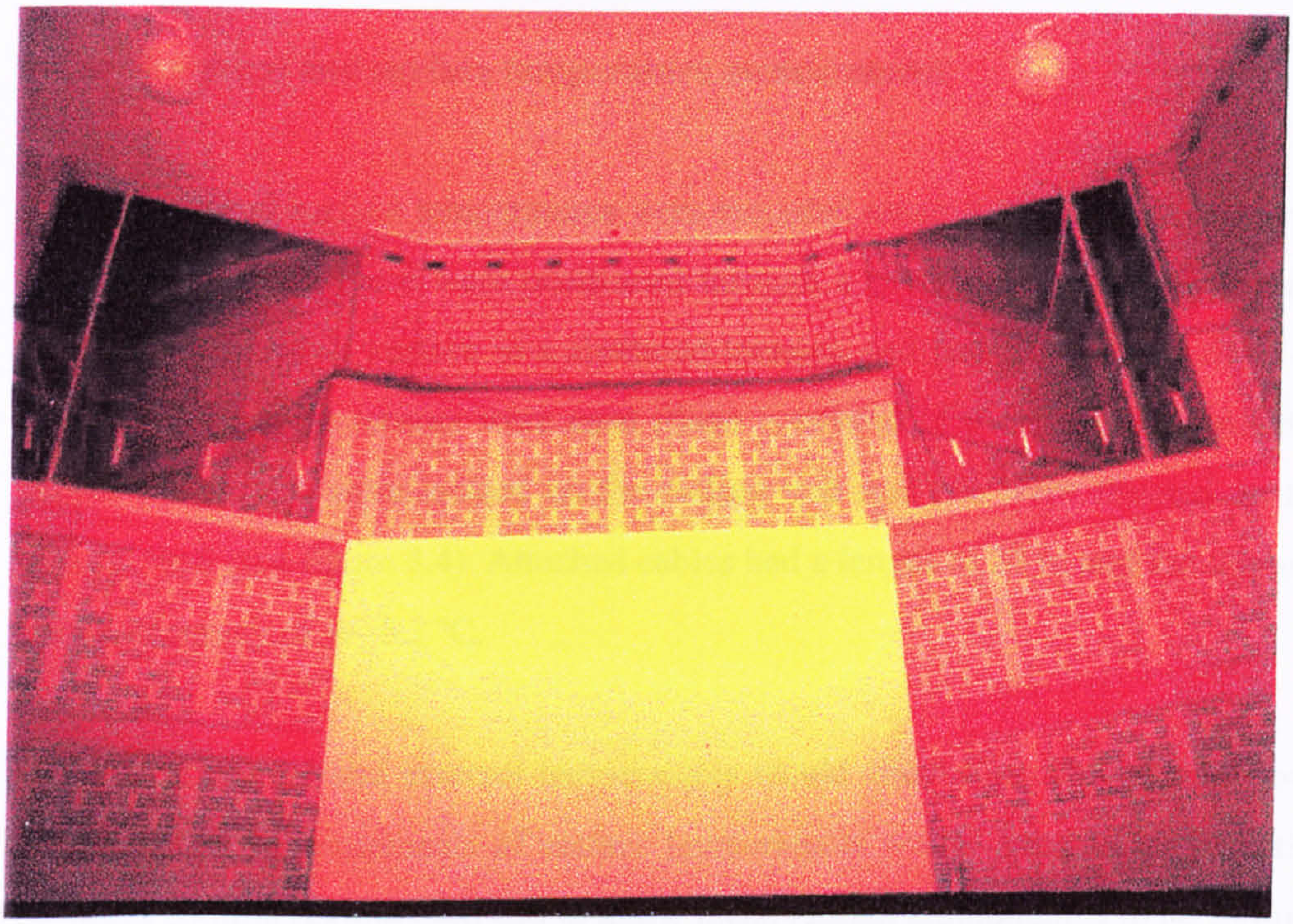


Figure 3.8 Stack inlets showing paper strips suspended across the openings

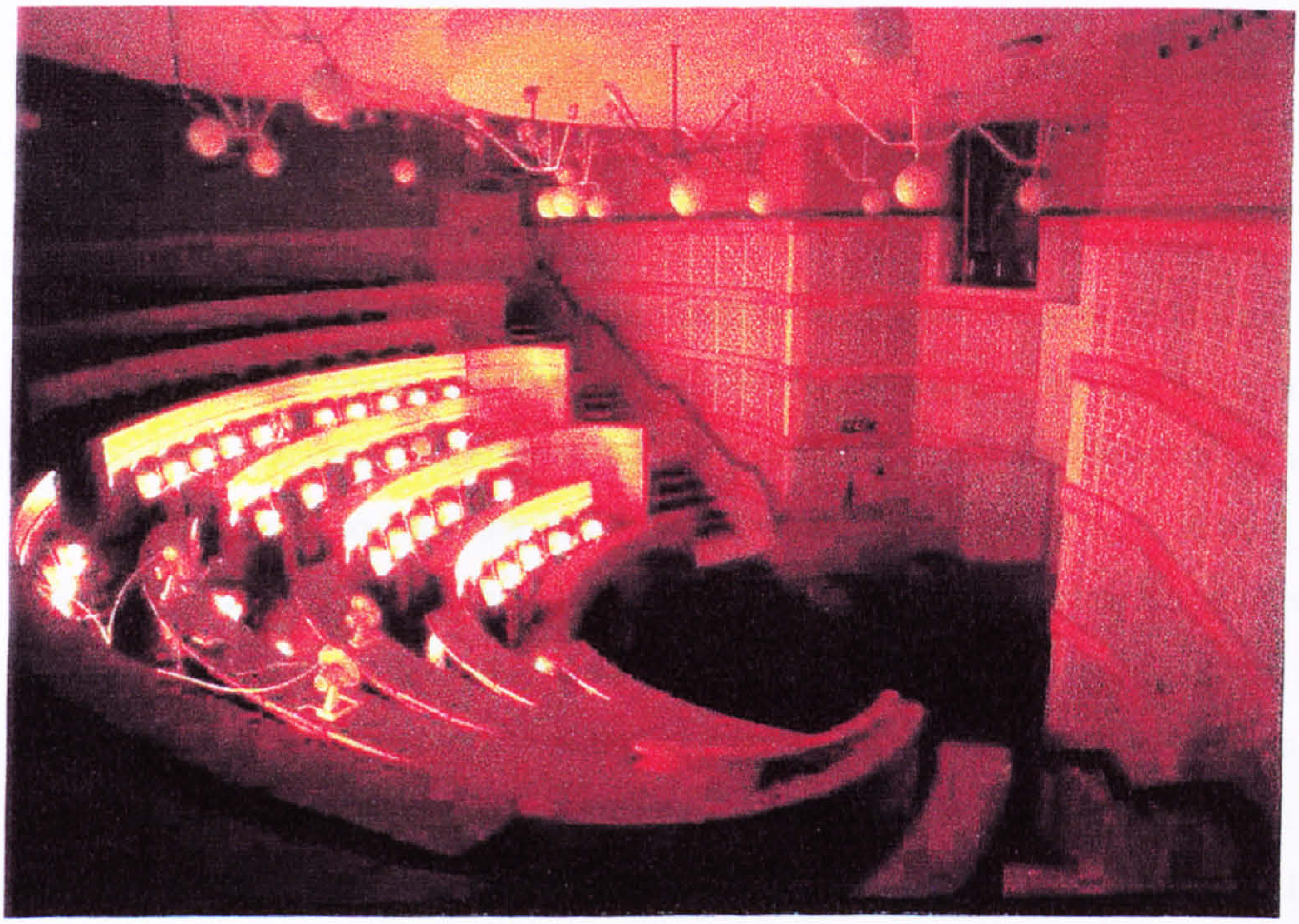


Figure 3.9 View of theatre seating with 100 W tungsten bulb heat sources

### 3.4.3. External and surface temperature measurement

External air and surface temperatures were recorded using 8 bit Squirrel loggers. Two air sensors were used, these were positioned through a north facing stairway window about 100 mm from a wall surface in a sheltered location and had an accuracy of +/- 0.1 °C. 4 surface temperature sensors were used (these consisted of thermistors mounted against a copper plate); two were located on the ceiling slab and two were situated on the curved internal wall (see figure 3.4). Attached cables had a length of 25 m and the output readings had an accuracy of +/- 0.2 °C.

Values for internal and external temperatures would allow buoyancy driven flow rates to be calculated. Measurements of surface temperatures would indicate the effectiveness of the night cooling regime used and to allow mean radiant temperatures to be estimated.
The use of a globe temperature sensor (to record mean radiant temperatures) would have required obtaining a representative value for the average room air speed. It was not possible to obtain the latter value due to the limited number of air speed probes available

### 3.4.4 Wind monitoring

The BMS system has a weather station which is designed to monitor and record outside air temperature, wind speed and direction. However, comparison of data obtained with that procured from another site showed no correlation between wind direction values at both sites was obtained so this data had to be discounted (see figure 3.11 below). The BMS system appeared to overestimate local wind speeds. The lack of correlation between wind directions at the two sites is believed to be due to the mounting height of the instrument mast. The mast is located on top of one of the concourse stacks but is about 1 m above the top of the chimney. Thus sensor readings are strongly affected by local wind turbulence. Another source of data was obtained at East Midlands Airport and gave general indications of wind speeds and directions in Leicester after correcting for sensor heights and location.

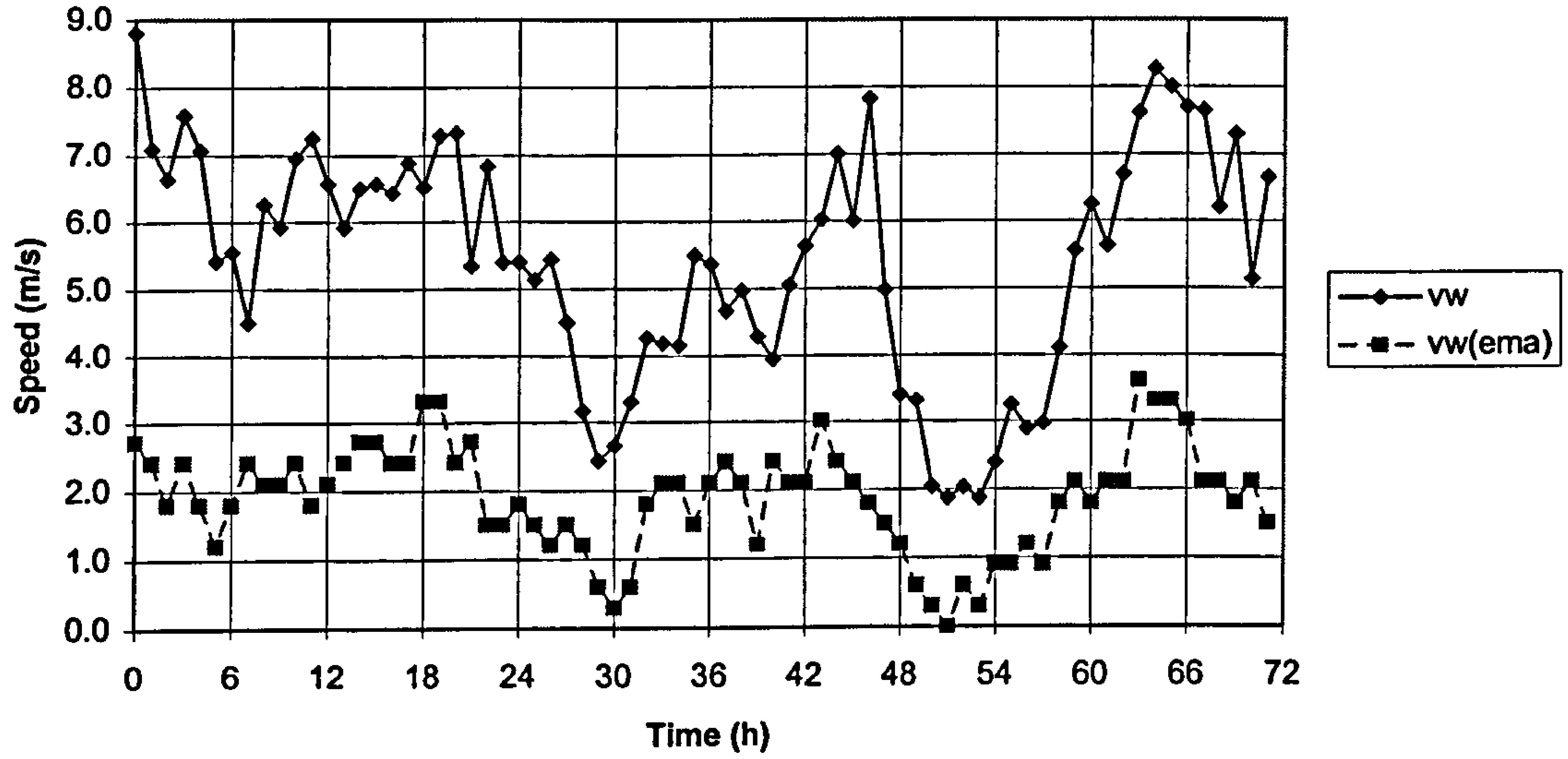



Figure 3.10 Comparison of wind speeds measured using the BMS and windspeeds recorded at East Midlands Airport.

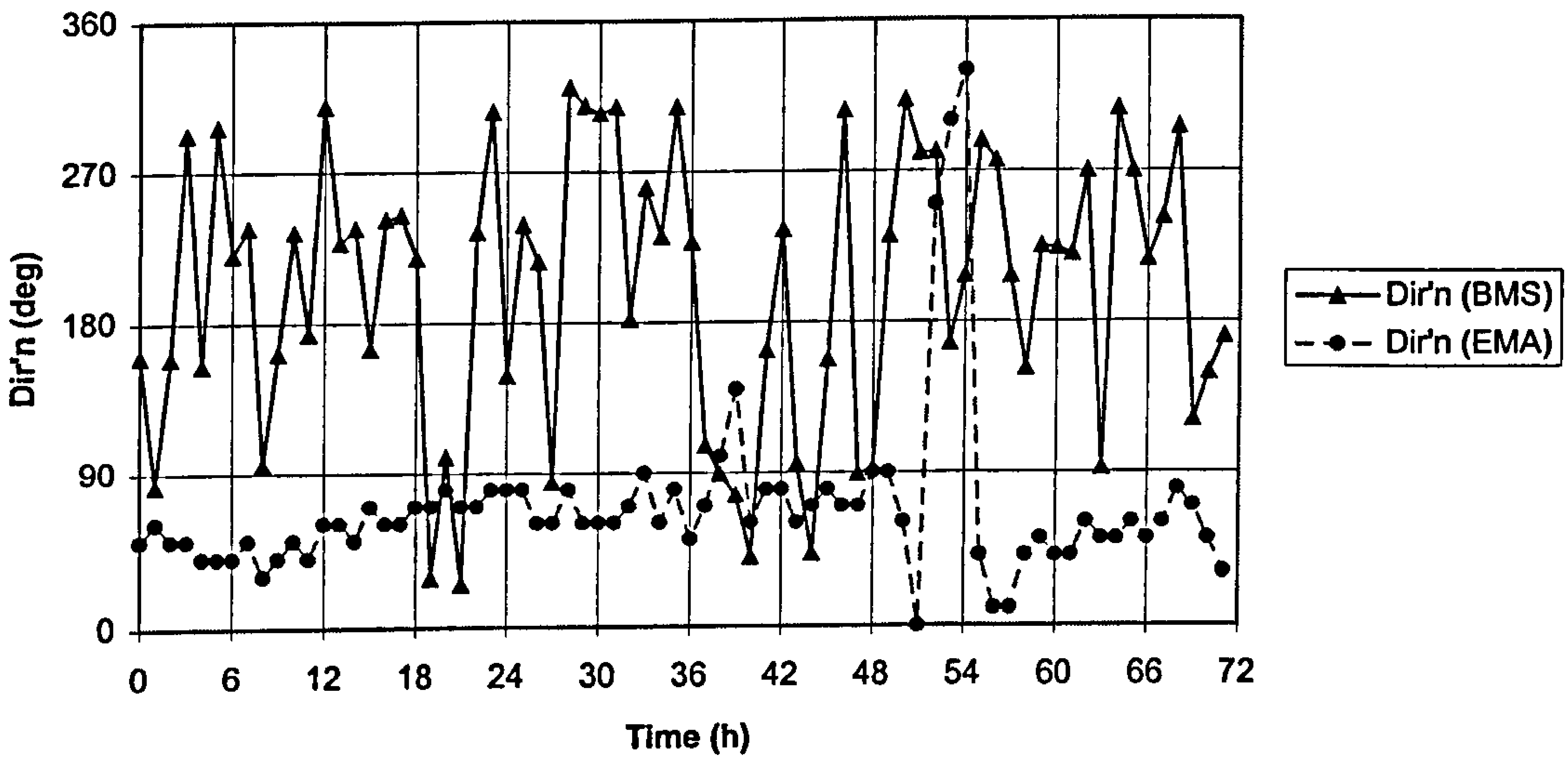



Figure 3.11 Comparison of wind directions measured via the BMS and at East Midlands Airport.

A dedicated weather station was erected on the roof of Gateway House, an adjacent building to the Queens Building and has been in continuous operation since September 1995. Data on wind speeds and directions is logged a 5 minute intervals and is down-loaded once a month using a portable computer. Data transfer takes approximately 10 minutes. This data correlates well with data obtained from a weather station located at Loughborough University (see figure 3.12).

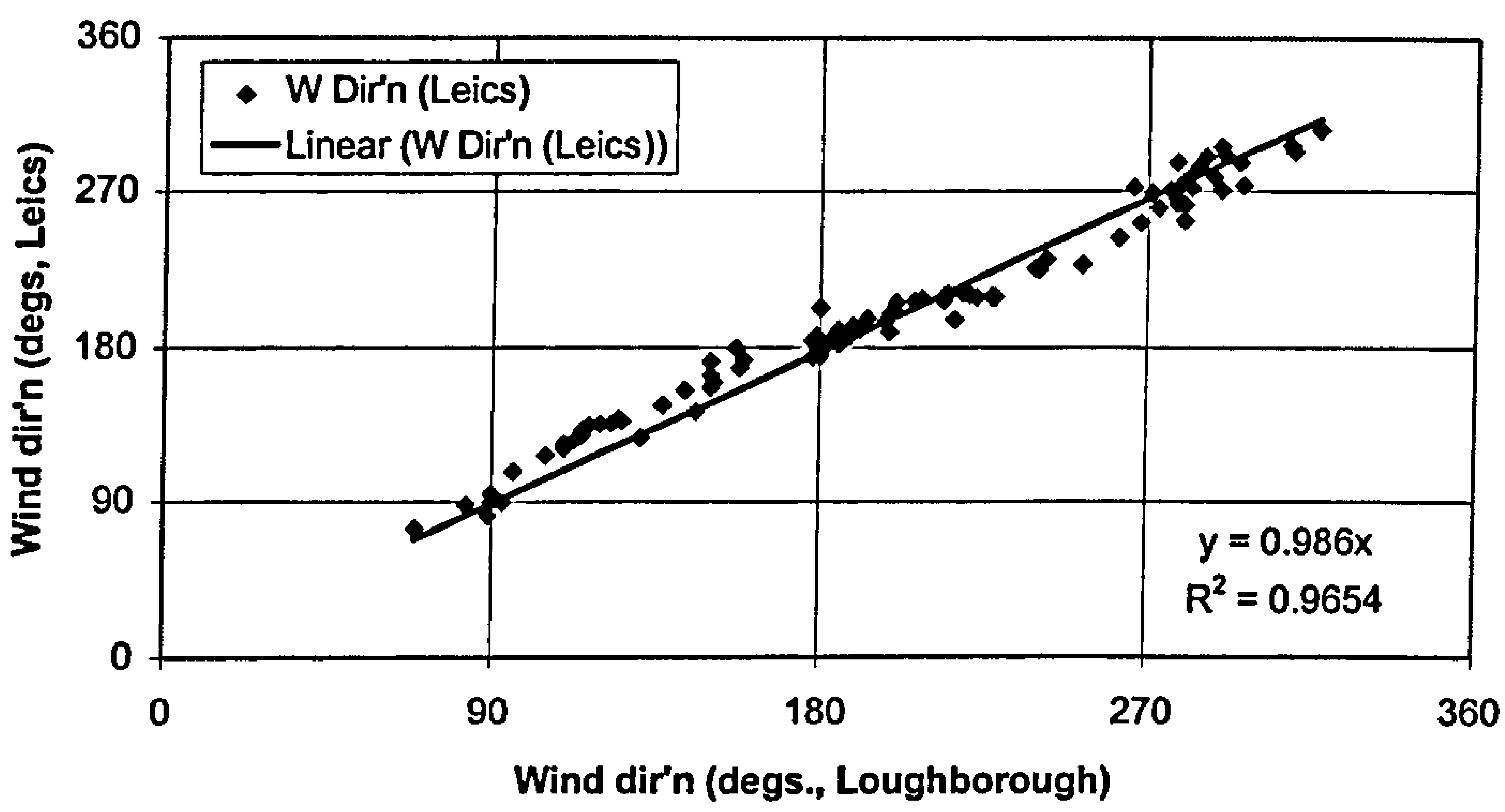



Figure 3.12 Comparison of wind directions measured at Loughborough University and at DMU, Leicester (November 1995).

## 3.5 Data processing

A computer program for changing the format of data obtained from the Autovent system had to be written so that the data (i.e. values of tracer gas concentrations and room temperatures) could be imported into a spreadsheet program and analysed. The language used was BASIC.

Data from the BMS was provided in the form of text files containing one column of data (i.e. the point name and its particular values obtained at 15 minute intervals over a 7 day period). Another program was written to generate output files containing several columns of data, in a format that would be compatible with a spreadsheet program.

### 3.5.1 Calculation of local and average air-change rates:

The local air change rate is found by dividing the initial concentration of tracer gas by the area under the concentration v. time curve; i.e.

Local air change rate ($ACR_p$) at point p

$$ACR_p = \frac{C_p(0)}{\int_{t=0}^{t=\infty} C_p(t)\,dt} \tag{3.1}$$

The average air-change rate ($ACR_{avg}$) in the room is then found by taking the average of the local air change rates at 6 locations ($n_{pts} = 6$):

$$\text{Average } ACR_{avg} = \frac{\sum_{i=1}^{n} ACR_i}{n_{pts}} \tag{3.2}$$

Ventilation effectiveness at a point is a measure of how well incoming fresh air mixes with room air at that point. This can be found from:

$$\text{Ventilation effectiveness at a point p} = \frac{ACR_p}{ACR_{avg}} \tag{3.3}$$

## 3.6 Experimental difficulties and how they were overcome

Note that the first sections in the following paragraphs outline the particular difficulties encountered while the second sections state how or whether the difficulties were partially or wholly overcome in practice.

1) It was difficult to achieve complete mixing of tracer gas before commencing an experiment. Mixing fans could only be used in a limited number of areas and could not be used at high level.

   Vents were opened (i.e. adequate mixing was assumed to have occurred) when differences of 20 ppm or less occurred between concentrations measured at 6 points.

2) The maximum values for measured air-change rates were greater than 20 thus very rapid decay rates were recorded. This necessitated using a high sampling rate.

   The normal sampling rate of 2 minutes was halved by connecting every sample point used to two channels on the sampling equipment but this meant that only six sampling channels could be used.

3) The length of sampling tubes used varied slightly from point to point. Ideally each tube should have the same length so the time taken for samples obtained at each point to reach the detector is the same.

   In practice, the average sample tube length was about 30 m and little deviation in length occurred.

4) The measurement of pressure differences across the stacks was attempted but this proved to be no easy task. Detected pressure changes are very small (less than 5 Pascals) but pressure values had a large standard deviation about their mean. Some method of averaging out rapid fluctuations in pressures was required.

It was decided not to continue with further pressure measurements as the sensors and associated apparatus was not sensitive enough to obtain meaningful readings.

5) Positioning of injection and sampling tubes and sensors was difficult due to the complex geometry of the space. The stepped nature of the seating layout and the large floor to ceiling heights made it difficult to gain access to the ceiling slab. It would have been useful if accurate sensors were installed during the fit-out of a building; this would facilitate the monitoring of the buildings performance over longer periods (it was found that a velocity sensor fitted into one of the stacks was unsuitable for measuring velocities below 2 m/s, was very inaccurate and of little use).

   Some tubes has been installed during the fit-out phase of the building; these were then extended beyond grill locations into the space. A number of high level tubes were positioned using a long plastic pole and mounted over lighting supports.

6) Injection of tracer gas could not be achieved when the space was occupied. Injection tubing plus fans needed to be placed in occupied areas. Hence $CO_2$ concentrations were measured instead. Thus some means of monitoring occupancy levels over extended periods (up to 6 hours) was required. This was achieved by counting heads. A more satisfactory method would be to use a camera with an internal timer and with a date/time recording facility and to take pictures at regular intervals (say 10 minutes after a lecture commences, once an hour).

   Most runs were completed while simulating average occupany levels. Runs carried out while the space was occupied involved measuring $CO_2$ concentrations while monitoring occupancy levels by counting the number of people present.

7) The minimum opening setting of the inlets and outlets gave a relatively large opening area; thus uncomfortable conditions (i.e. air speeds above 0.5 m/s and inlet temperatures less than 10 °C) could be produced. This made it difficult to carry out

ventilation measurements when the theatre was occupied and when the outside temperature was low. In practice, the vents would remain closed in winter.

Measurements of room air speeds were carried out when the space was unoccupied.

## 3.7 Performance of experimental methods

### 3.7.1 Tracer gas decay method:

1) The decay method and equipment used allowed the relative fresh air distributions on either side of the room, in the occupied zones and at high level to be found. Local air change rates due to infiltration could be estimated more accurately than those generated when the vents were open (with effective areas greater than 2.43 $m^3$). The method became too inaccurate when air change rates exceeded c. 20 (above which direct measurement of stack velocities should be used to find global flowrates).

2) The accuracy of the tracer gas decay method is +/- 10 % [8].

### 3.7.2 Velocity measurement:

1) Ventilation flow rates could be estimated more accurately if the air velocities in the stacks exceeded c. 0.3 m/s. The manufacturer of the low velocity Dantec sensors gives a % error of less than +/- 10% at velocities above 0.25 m/s (flows perpendicular to transducer axis). However the errors due to the nature of the flow, i.e. degree of flow turbulence and flow profiles, are likely to be more significant. Above 0.3 m/s, flows were likely to be less turbulent and more unidirectional.

Local air flow directions were found using a hand-held smoke generator while the direction of air movement into and out of stacks was detected using paper strips suspended across the stack inlet openings.

2) A total of 24 low velocity sensors was available. However, the distances of the sensors from the recording and logging equipment meant that only 15 sensors could be used in practice (two lengths of cable were required for sensors mounted adjacent to inlet grilles on the fifth row and in the rhs stack). Thus only 7 sensors were available for measuring room air speeds. The cables available had a length of 30 m and the distance between the logging/recording apparatus for several of the points used was greater than this distance. Hence more than one cable was needed to measure air speeds at these positions.

3) It was difficult to mount and access sensors positioned in the stacks. Rigid sensor supports were required and temporary cable supports were needed.

# 4 Results

## 4.1 Summary and objectives

Chapter 4 will detail the various results achieved during "winter", "mid-season" and "summer". Results have been obtained for bulk flowrates, air speeds, air and surface temperatures, and infiltration rates for different opening configurations (i.e. inlets and outlets open or closed). Data is presented in summary form to aid assimulation. Figure 4.1 gives an overview of the experimental configurations used.

The objectives were to :

- Find the key effects affecting ventilation rates through the enclosure and comfort conditions in the space. These would be achieved by measuring local and global air change rates; air velocities within the stacks and at critical points within the space (i.e. at locations where high velocities were likely to occur);

- Determine the nature of the ventilation system (is it similar in nature to a displacement or a mixed flow system?); this would be achieved by measuring the variations of temperatures with height and by using visualisation techniques;

- Calculate the ventilation effectiveness (using data obtained from the above);

- Compare the results of the tracer gas and velocity measurements with those obtained using zonal models (e.g. ESP).

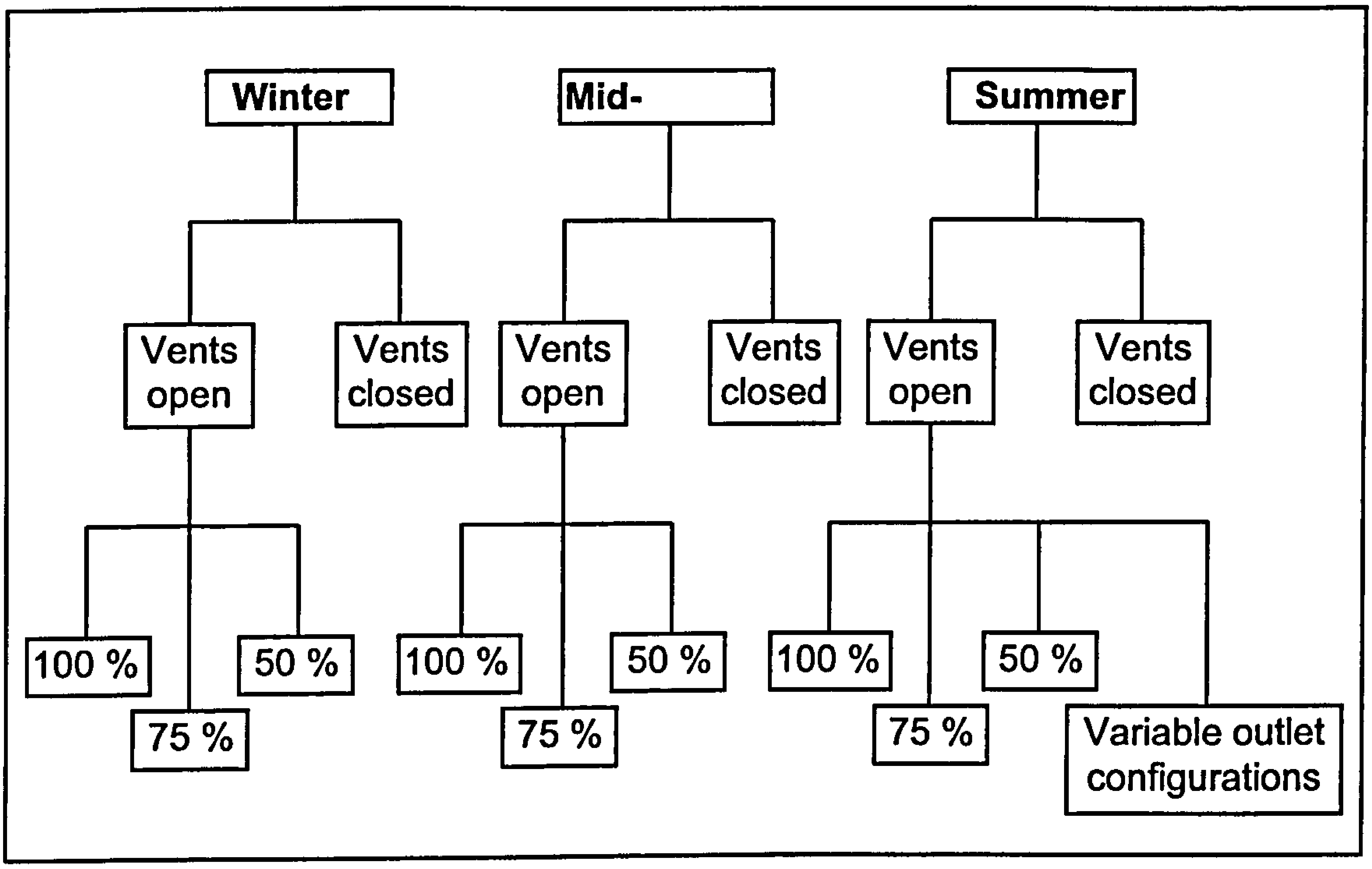



Figure 4.1 Experimental configurations used

## 4.2 Chapter format

The chapter starts with general discussions on thermal comfort parameters, indoor air quality (IAQ), bulk flowrates and the experimental configurations used throughout the experimental period.

The next sections present in summary form the results obtained in "winter", "mid-season" and "summer" for the vents in closed and open positions. "Winter" is deemed to occur when the average external temperature is 5 °C or less, "mid-season" conditions apply when the external temperature is about 10 °C, while "summer" conditions occur when the external temperature is at or above approximately 15 °C. Results presented in these sections include internal air and surface temperatures, and air speeds at selected positions in the room, or conditions generally relating to thermal comfort parameters.

Information on indoor quality (IAQ) is then presented for the three seasons. IAQ is deemed to depend largely on carbon dioxide concentrations. Adequate IAQ is assumed to occur if the $CO_2$ concentrations are maintained below 1,000 ppm.

The next section deals with bulk flowrates or total flowrates through the space for a range of external conditions. It is shown that the CIBSE stack formula (given in detail later) gives good predictions of flowrate using the inside and outside air temperatures and an accurate value for the effective vent area. Some data is also presented on infiltration rates.

Data on average and local air change indices are then presented, indicating the effectiveness or otherwise of the ventilation system used. Information of 6 zones within the space is supplied.

A section on air movement in the room describes the general direction of air flow and gives an explanation for the phenomenon of downdraughts issuing from the stack inlets when the outlet vents commence closing.

Some information is provided on environmental conditions in lecture theatre 1.12 (which is adjacent to 1.10 and has been described in chapter 1). Results given here were obtained by general observation and from spot measurements.

The last section of this chapter examines the effects of varying the outlet configurations (i.e. with specific vents open and others closed) to see if higher flowrates could be produced with leeward vents open and windward vents closed.

## 4.3 Thermal comfort parameters

Thermal comfort in an enclosure depends upon a range of variables; these can be separated into (a) environment variables and (b) occupant variables. The most important environment parameters (for occupants in the auditorium) are inlet air velocity and temperature and mean radiant temperature. It is reasonable to assume that little variation in air temperature occurs horizontally or vertically in the occupied zones. It is also assumed that opposite surfaces have the same or similar temperatures (i.e. no radiant temperature asymmetry exists). Relative humidity values are assumed to lie between 40 and 70 %. Occupant variables are "Clo" values (relating the levels of clothing worn) and activity levels. These latter parameters can be assumed to be constant.

Two distinct "zones" of interest can be defined; the seating area occupied by the audience, and the front of the lecture theatre where the lecturer is located.

The primary concern is the determination of comfort conditions adjacent to inlet grilles (i.e. local comfort levels, at locations where air temperatures and speeds are likely to be critical in winter and mid-season, when the vents are partially or fully open) and at the front of the theatre (where stratification of colder air can occur). Air speeds drop off rapidly away from the grilles and air temperatures are generally 18 to 22 °C in other areas. So the problem reduces to: what air speeds and temperatures occur at ankle height opposite the inlet grilles for different opening settings, and over what range of values are occupants likely to be uncomfortable? Conditions at the front of the theatre also need to be explored. Other variables (e.g. surface temperatures) are likely to be irrelevant for seated occupants if the above assumptions are accepted (but will have some effect on the lecturers comfort levels).

A scale from -3 to +3 can be assigned to particular combinations of air speeds and temperatures. Detailed calculations, using PMV (predicted mean vote) formulae, are not required due to the limited number of critical variables involved (i.e. assuming constant activity, "clo", and humidity levels) [13].

In practise, only a relatively small area of the theatre is "critical", i.e. where inlet grilles are directly in line with external inlets. In these areas, cold air can enter directly into the space in winter and in mid-season. When this occurs, very uncomfortable conditions are likely thus a PMV score of -3 would be assigned in these areas. Cold air temperatures can also be observed at the front of the space, where there is strong evidence of temperature stratification.

## 4.4 Indoor air quality

$CO_2$ concentrations are used as a primary measure of indoor air quality (IAQ) in the auditorium. A lower absolute limit commonly used is 600 ppm and a recommended upper limit is 1,000 ppm (parts per million) of $CO_2$, however the safety limit is taken to be 5,000 ppm [14]. Unconsciousness may occur when the concentration exceeds 100,000 ppm (10 % by volume). The external concentration is approximately 350 ppm; values in excess of this may be experienced in very polluted environments, e.g. areas adjacent to heavily used major roads.

The control system is designed to respond to $CO_2$ levels; if the level reaches 1,000 ppm then the vents will fully open, provided certain conditions are satisfied (see the section on **building controls**, p. 6).

Local air change rates were found by injecting $SF_6$ gas and measuring the decay rates of the gas concentration at 6 points in the auditorium. This gave a measure of how well fresh air was distributed in the space.

The infra-red gas analyser used could simultaneously measure $CO_2$ and $SF_6$ concentrations at selected points. $SF_6$ was used when the space was unoccupied; $CO_2$ measurements were used when the room was occupied. A number of runs were carried out with the space partially occupied (c. by approximately 50 people). Some plots of $CO_2$ concentrations versus time are in figures 4.13 and 4.14.

$CO_2$ levels were used to measure infiltration levels when the space was unoccupied. The decay technique was used.

Other indications of IAQ are dust concentration in the room air and the levels of dust accumulation within the underfloor void and at the base of the stacks. Some dust accumulation was observed within the underfloor plenums but systematic measurements of dust volumes were not performed.

Indoor humidity levels were not monitored for extended periods. However the indications are from spot measurements that humidity levels do not vary greatly from morning to evening (c. 55%) and may only become elevated after extended periods of high occupancy levels (i.e. greater than 100 occupants).

## 4.5 Bulk flow rates

Bulk flow rates through the theatre are driven by pressure differences generated by temperature differences between inside and outside as well as wind effects. In winter, temperature differences are of the order of 15 °C and stack effects predominate. In mid-season (average temperature difference is 10 °C) flows are also generally driven by stack effects but wind driven flows are becoming significant. In summer flows can be driven entirely by wind effects (i.e. flow rates are generated even when the average inside temperature equals the external air temperature). However, it is difficult to generalise for naturally ventilated buildings in a UK climate; wind driven flows can dominate in any particular season provided wind speeds are above (say) 10 m/s. Tables (see Appendix A) have been produced which summarise conditions in the space. Included with this data is a general indication of wind strength. Low wind speeds occur between 0 and 2.7 m/s; medium wind speeds occur between 2.7 and 9.5 m/s and high wind speeds occur above 9.5 m/s. These ranges are based on the Beaufort scale. Airchange rates have been plotted against a function of inside/outside temperature differences in figure 4.18.

The above of course assumes that the vents are partially or fully open (i.e. flows can only occur when the vents are open). When infiltration rates are measured, there does not appear to be any significant correlation between flow rates and inside/outside temperature differences.

Flow rates were found from tracer gas measurements and from velocity readings obtained in the two stacks. A comparison between the two methods of finding flow rates is shown in figure 4.17.

## 4.6 Experimental configurations

### 4.6.1 Initial experimental configuration

Measurement points used are as indicated in figure 3.4. In all, 2 injection points (split two ways), each located on one side of the theatre and 12 sample points were used for initial runs (up to the end of 1994). The injection points were placed adjacent to desktop mixing fans on the fourth row from the front of the theatre while sampling points were positioned in the following areas: i) stacks; ii) at the front of the room at a height of about 2 metres; iii) on the left-hand side (lhs)and right hand side (rhs) near the front row grilles; iv) on the lhs and rhs, fourth row up; v) at high level, about 1 metre below the ceiling slab, on the lhs and rhs; vi) at high level, at the mid-section of the space; and vii) within the inlet plenums.

A thermistor probe was attached to sampling points i) , ii), v), and vi) above to allow "zone" air temperatures to be recorded.

### 4.6.2 Final experimental configuration

The layout of sampling points was similar to that used for the initial runs, except that no tubing was placed within inlet plenums and only six samples were fed into the infra red gas analyser (i.e. sampling tubes from the stacks, low level and mid-level positions on the lhs and rhs were combined such that samples could be obtained every minute). See figure 3.4.

The revised configuration allowed the tracer gas sampling time interval to be halved so that twice as many measurements could be made over a given experimental period.

## 4.7 Thermal comfort parameters

### 4.7.1 General comments

Due to the location of inlets and outlets the local air change rates were generally significantly higher on the right hand side (rhs) of the theatre (than on the left hand side (lhs)) (inlet openings are positioned on the rhs of the space) (see figures 4.22, 4.23, and 4.24 ). There did not appear to be large differences between flow rates through the lhs and rhs stacks (see figures 4.4 and 4.7).

The temperature differences between low and high level zones were generally about 2 °C, with little variation occurring with changes in flow rates. The depth of the warmer high level zone appeared to decrease with increase in air-change rate, and greater temperature fluctuations were observed at high level than at low level (i.e. c. 0.5 °C and 0.2 °C respectively).

PMV (predicted mean vote) values can be found by calculation from Fangers equations [13,15]. PMV values should be viewed with caution and give a general indication of comfort levels only. They have been derived from experiments in mechanically ventilated rooms where conditions were closely controlled, rather than in naturally ventilated spaces with highly turbulent air flow regimes (as is the case for the De Montfort Auditorium).

### 4.7.2 Winter

The following conditions were recorded for "winter" external conditions (i.e. average air temperatures at or below 5 °C) [16]:

#### 4.7.2(a) Inlets and outlets closed

1) Stable room air temperatures occurred (19 to 21 °C) (due to the thermal heavy-weight nature of the building). The thermal mass of the space also kept air temperatures at or above 17 °C.
2) Ceiling slab temperatures showed less variation than room air temperatures. but behaved in a similar fashion (see figures 4.2 and 4.3).
3) Low air velocities: Generally inlet and room air velocities not directly opposite an inlet plenum have been low (less than 0.3 m/s on average).

Note that the drops in air temperature at 7:00 (see figures 4.2 and 4.3) are caused by the vents opening. This was due to an error in the original control algorithm. The heating valve was opened for an extended period during the pre-heat period (7:00 to 9:00 hours); it was opened for shorter periods during the daytime.

The vents were opened during experimental periods (see figure 4.3). Corresponding drops in air temperature occurred.

The low decay rate in air and surface temperatures is evident from figure 4.3. The biggest drop in temperature occurred after an extended holiday period, with a smaller drop occurring over a weekend. Hence the pre-heat time at the start of a week was greater than that occurring at the end of the week.

#### 4.7.2(b) Inlets and outlets open

1) For one particular run, average room air temperatures only dropped to 18 °C in spite of an external air temperature of 4 °C (see figure 4.5).
2) See 2) above, but greater variation in temperatures occurred.
3) Very low inlet air temperatures together with high air velocities were recorded (e.g. down to 7 °C with speeds above 0.5 m/s with external air temperatures of 3-4 °C) (much lower air temperatures were measured on the rhs).

It is evident that little or no ventilation was being provided during occupancy hours (i.e. there are no sharp drops in internal temperature, apart from at 7:00). Occupancy levels were zero or low, however the $CO_2$ sensors installed were mal-functioning so ventilation was being controlled using average temperature only.

Two way flow in the stacks was detected from stack temperatures (see figure 4.4). There was an increasing air temperature in one stack with a corresponding decrease in the other stack. The explanation for these conditions is outlined in section 4.11. The conditions take place when the stack vents begin to close (air enters into one stack vent and exits via the second stack vent) and occur over a period of several minutes.

A more gradual rise and decay of average air temperatures can be seen in figure 4.5 (compared to the changes in stack temperatures). Again the thermal mass of the building is having a significant effect on reducing temperature variations.

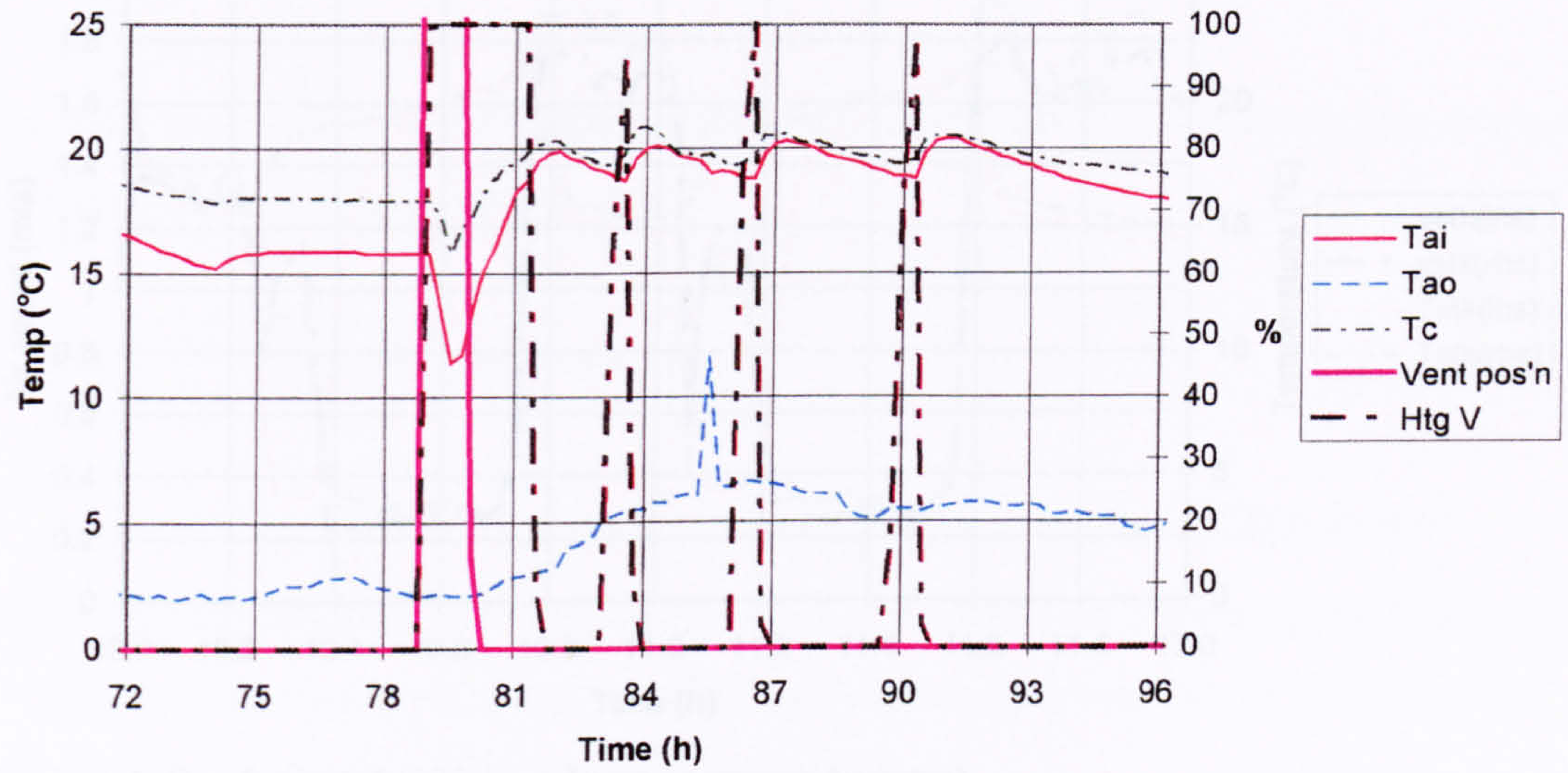



Figure 4.2. Internal and external temperatures (winter) over a 24 hour period.

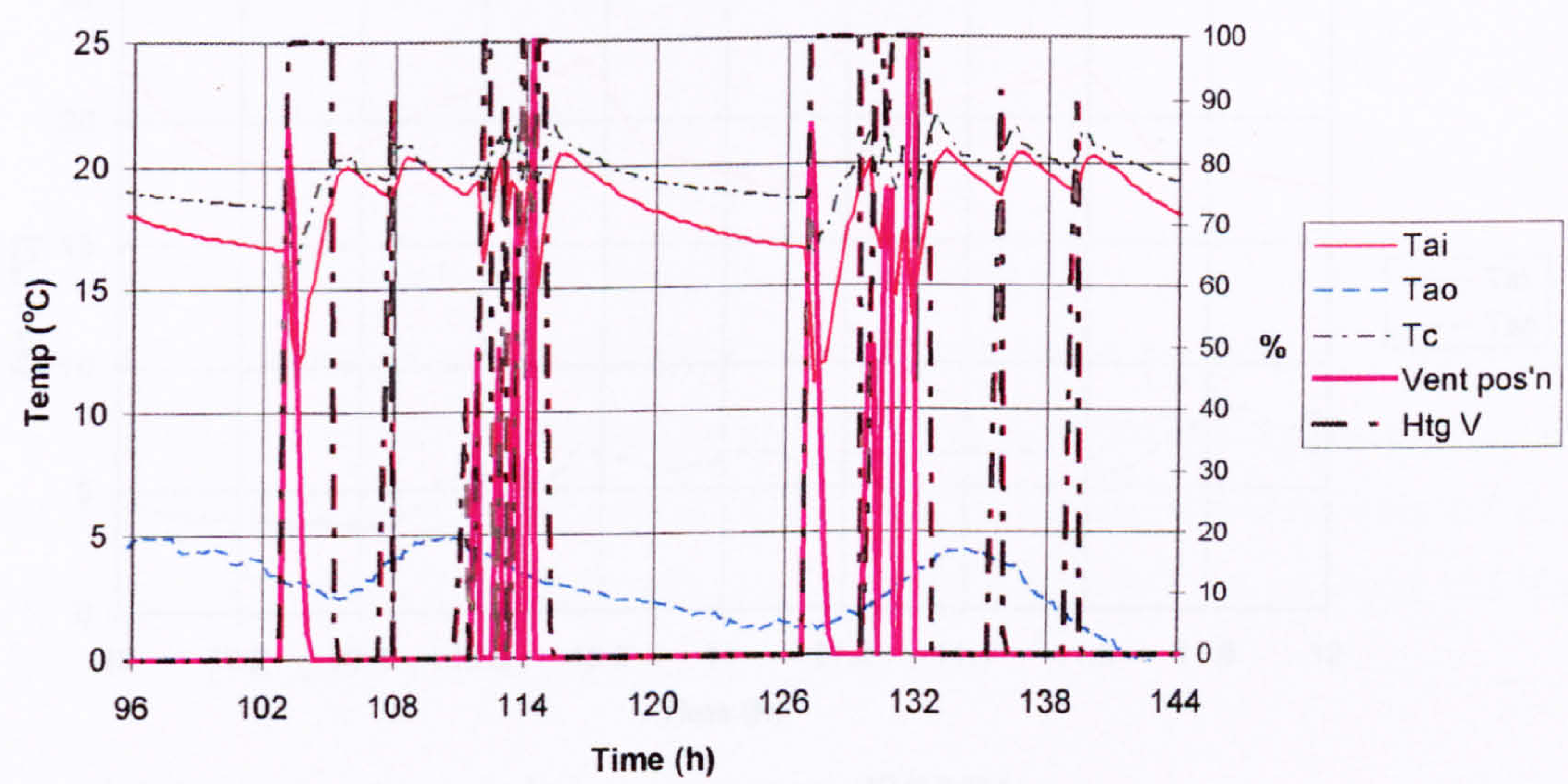



Figure 4.3. Internal and external temperatures (winter)

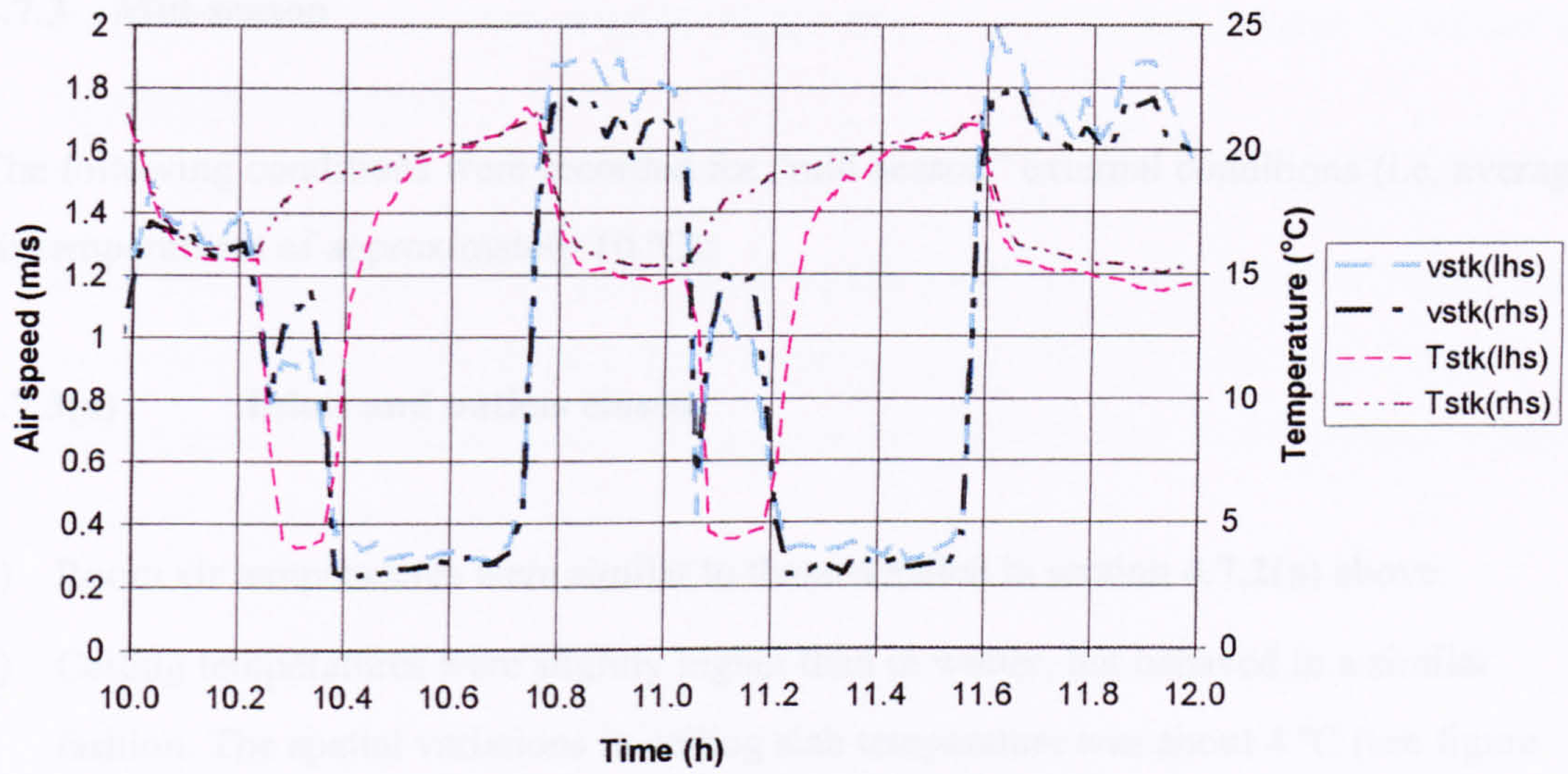



Figure 4.4. Stack air velocities and temperatures (winter)

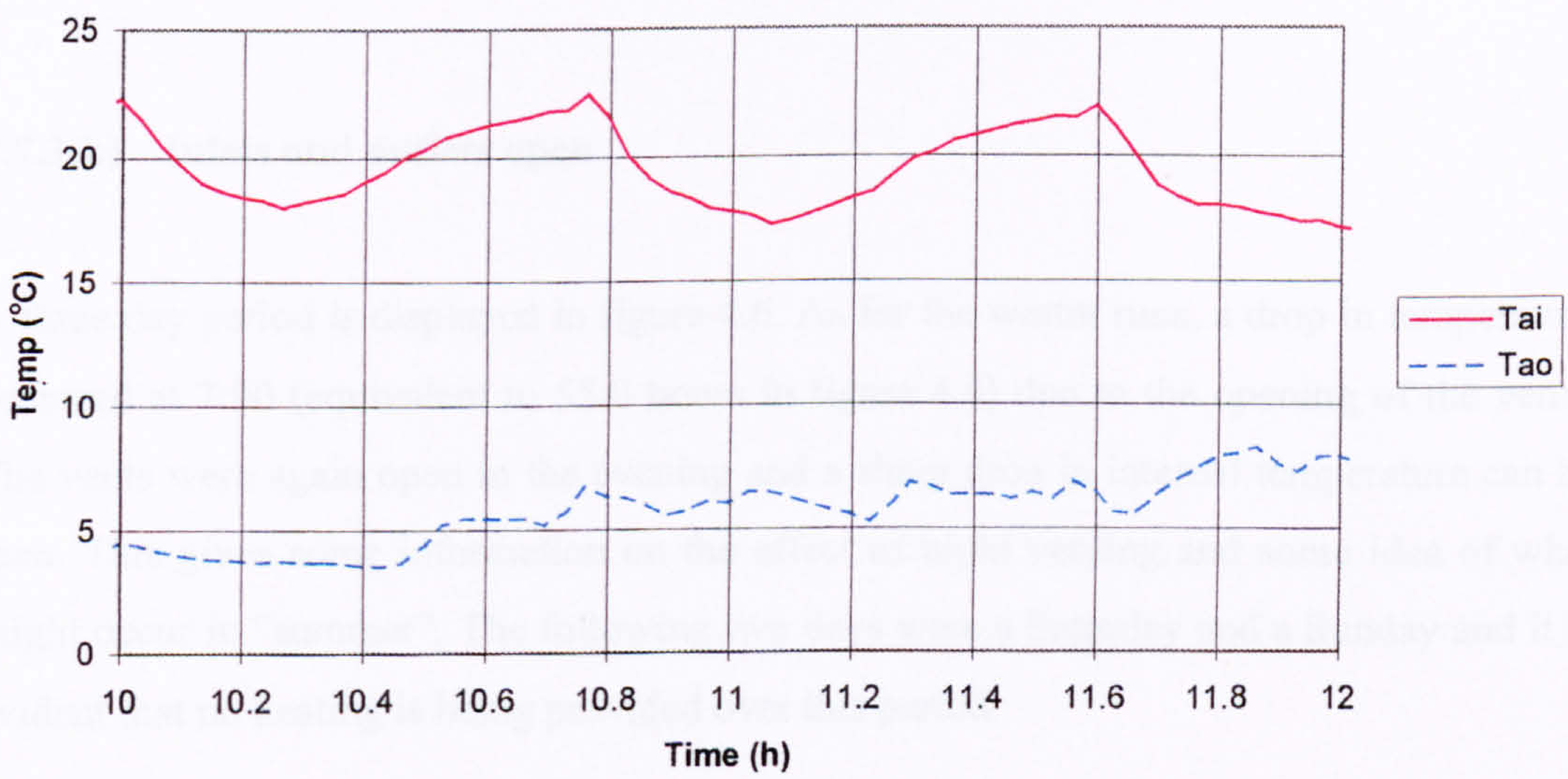



Figure 4.5. Internal and external air temperatures, 22/12/94

### 4.7.3 Mid-season

The following conditions were recorded for "mid-season" external conditions (i.e. average air temperatures of approximately 10 °C):

#### 4.7.3(a) Inlets and outlets closed

1) Room air temperatures were similar to those detailed in section **4.7.2(a)** above.
2) Ceiling temperatures were slightly higher than in winter, but behaved in a similar fashion. The spatial variations in ceiling slab temperature was about 4 °C (see figure 3.5 for locations of surface temperature sensors).
3) Low air velocities: Generally inlet and room air velocities were low (less than 0.3 m/s on average).

#### 4.7.3(b) Inlets and outlets open

A three day period is displayed in figure 4.6. As for the winter runs, a drop in temperature occurred at 7:00 (equivalent to 55.0 hours in figure 4.6) due to the opening of the vents. The vents were again open in the evening and a sharp drop in internal temperature can be seen. This gives some information on the effect of night venting and some idea of what might occur in "summer". The following two days were a Saturday and a Sunday and it is evident that no heating is being provided over this period.

The vents were open for short periods on the first and second days. An "exploded" view of the effects of opening the vents is shown in figure 4.7. Here stack air speeds and temperatures are displayed, with the speeds increasing as the vent areas are increased (going from left to right). Two way flow occurred as before (in "winter") while the vents were closing.

1) Ceiling temperatures were slightly higher than in winter, but greater variations in temperatures occurred.

2) Low inlet air temperatures were found on the right hand side (slightly higher than external temperatures) coupled with relatively high inlet air velocities; hence very uncomfortable conditions were produced (i.e. conditions which would produce a PMV value of -3.0) (see figure 4.9).

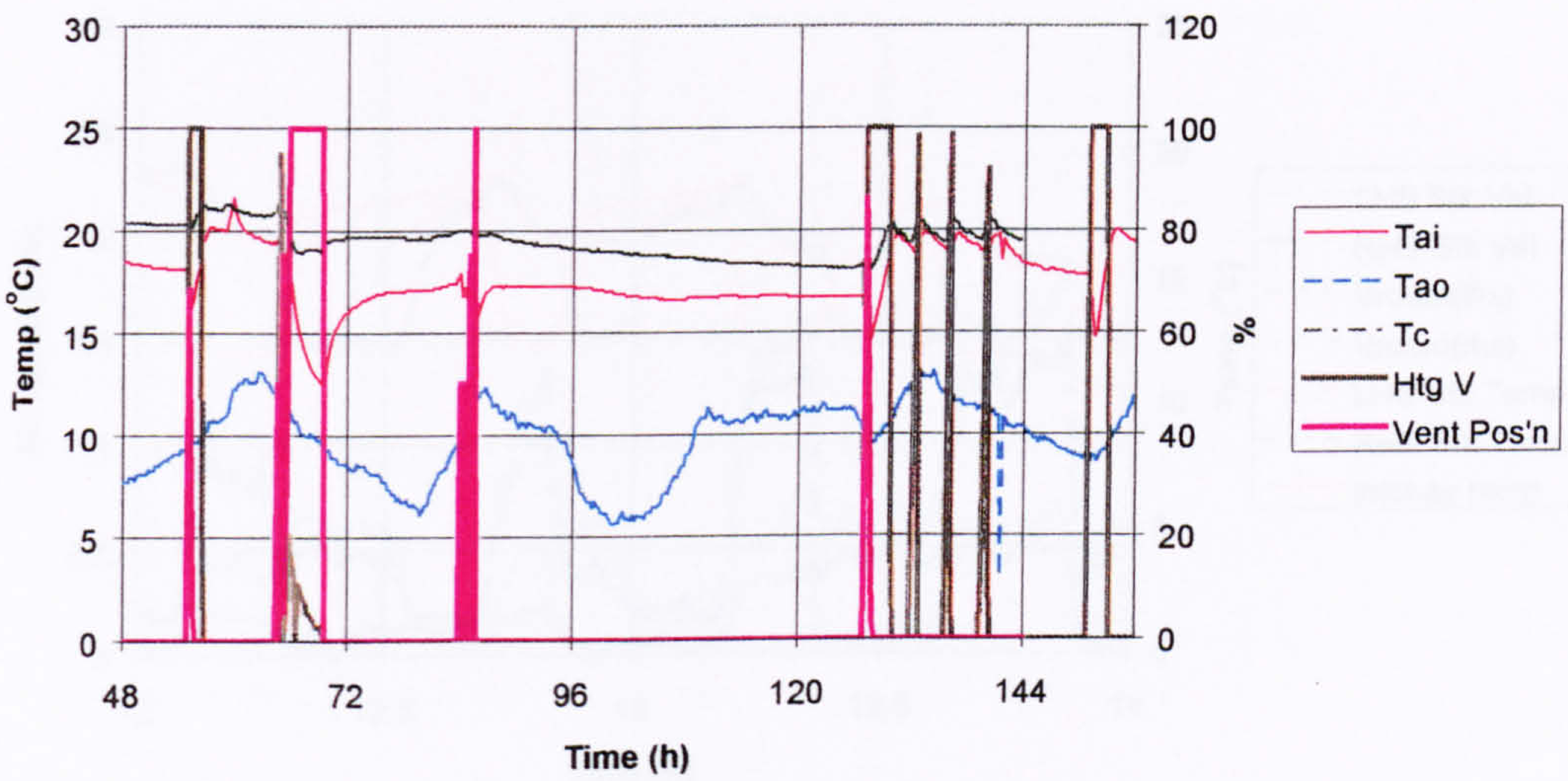



Figure 4.6(a) Internal and external air temperatures (3 & 4/2/95)

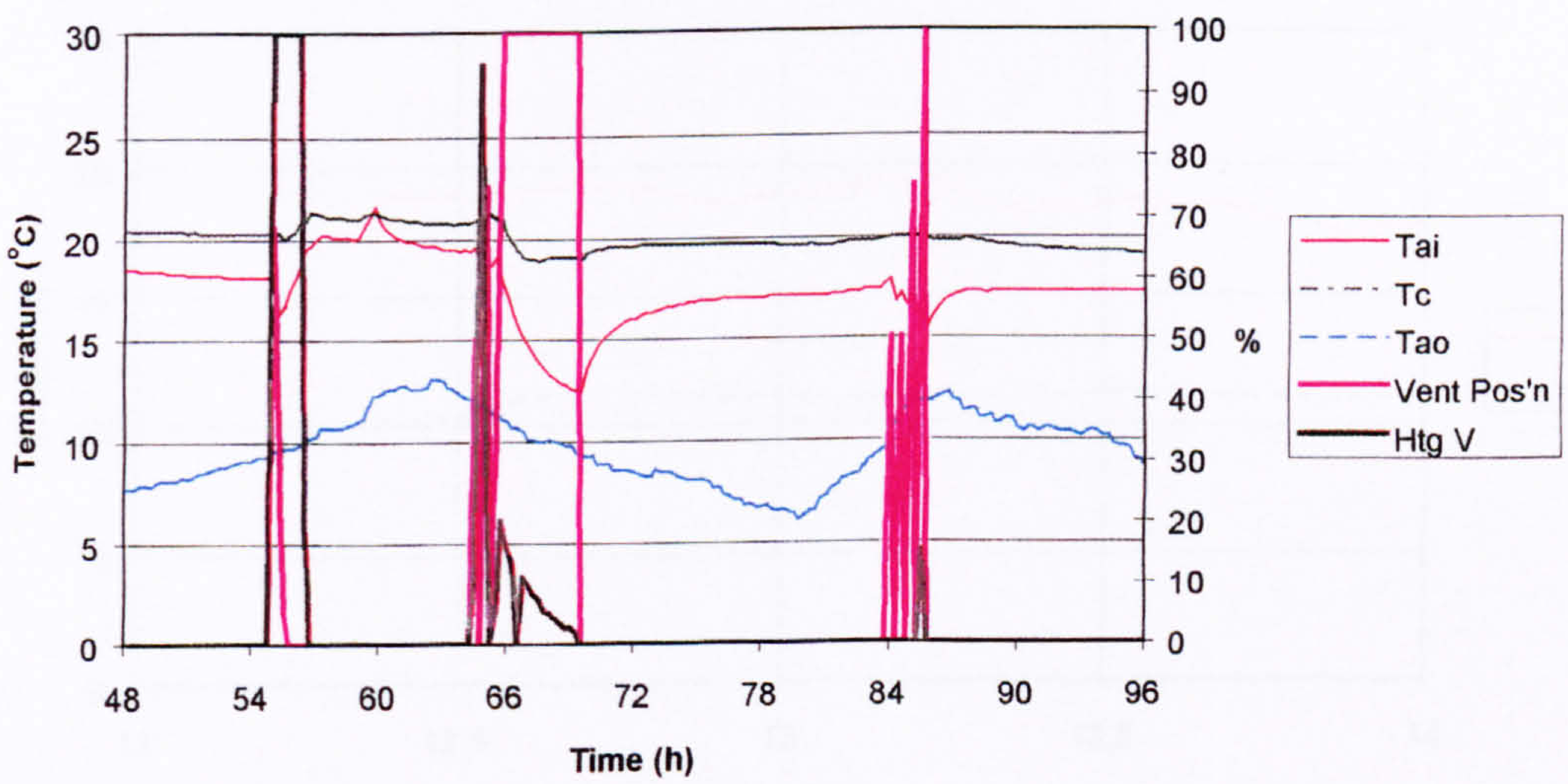



Figure 4.6(b) Internal and external air temperatures (3 & 4/2/95)

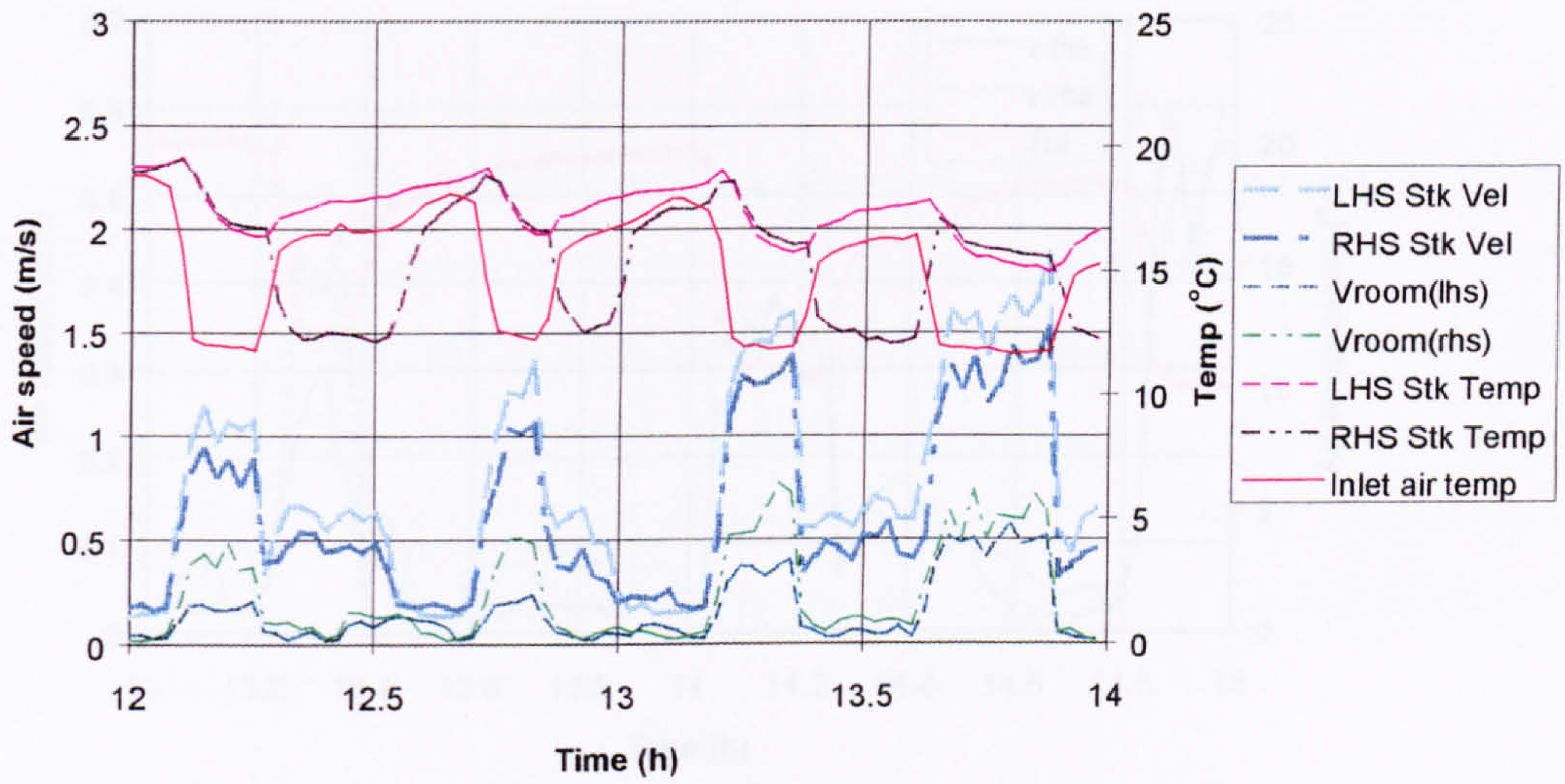



Figure 4.7 Internal and stack air temperatures (mid-season)

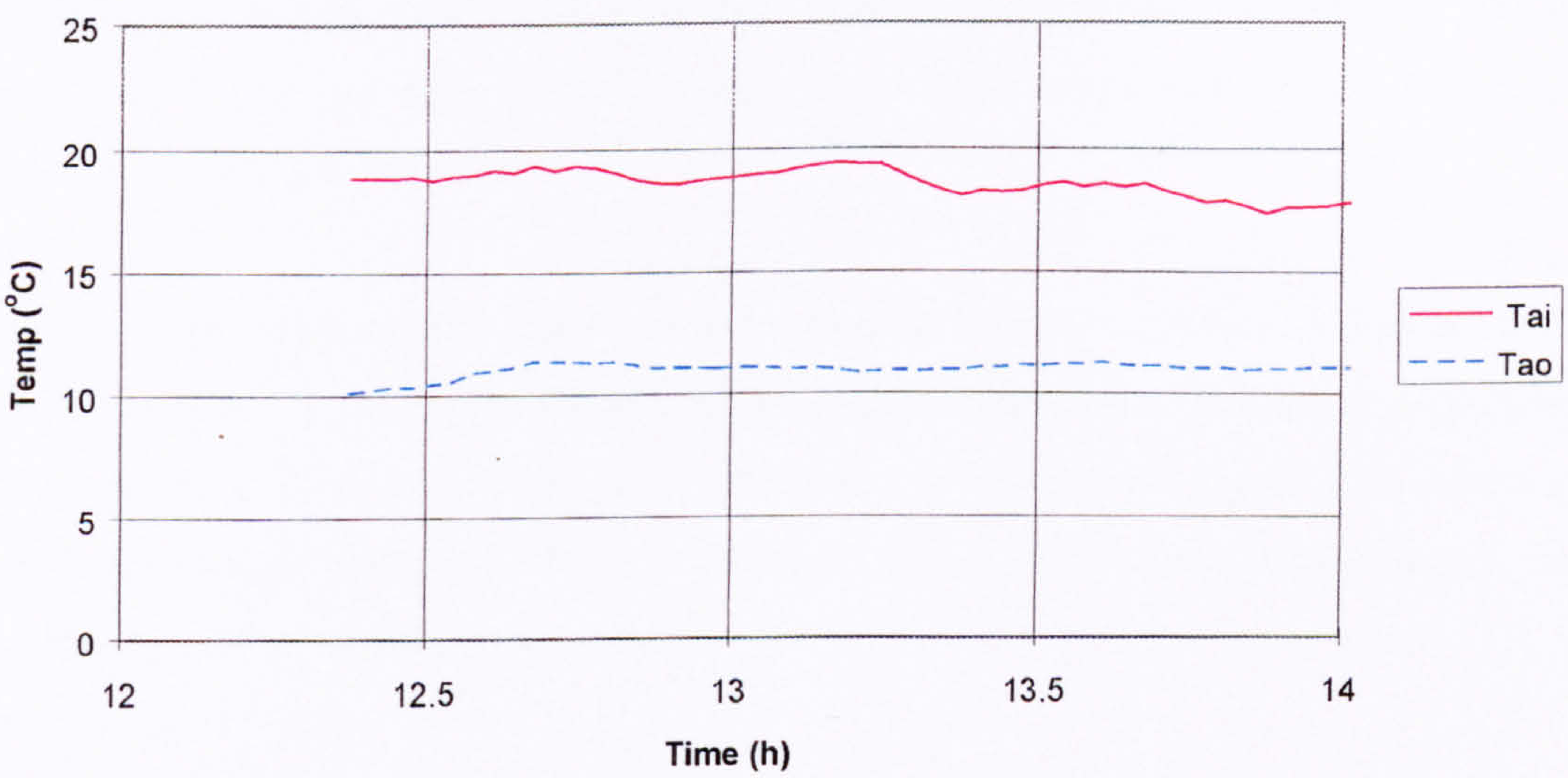



Figure 4.8 Internal and external air temperatures (mid-season)

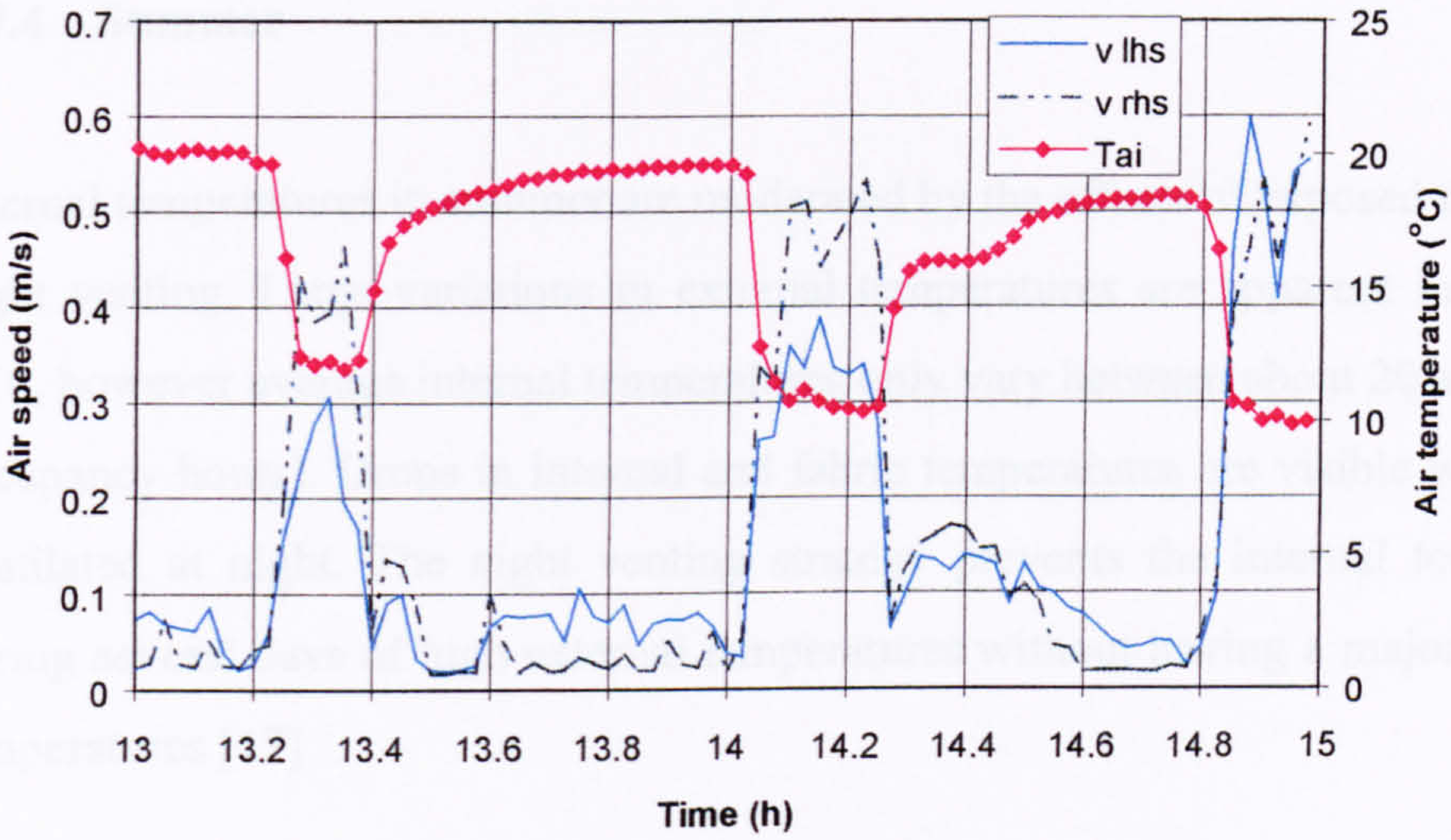



Figure 4.9 Room air speeds and internal air temperatures (mid-season)

### 4.7.4 Summer

Internal temperatures in summer are moderated by the effects of exposed thermal mass and night venting. Large variations in external temperatures are apparent in figures 4.9 and 4.10, however average internal temperatures only vary between about 20 and 23 °C (during occupancy hours). Drops in internal and fabric temperatures are visible when the space is ventilated at night. The night venting strategy prevents the internal temperature rising during several days of high external temperatures without having a major effect on fabric temperatures [17]

The control system causes the stack fan to operate a few hours after mid-night; this appears to produce an un-wanted rise in the room temperature (possibly due to de-stratification of the air in the space). It is likely that the fan will not be very effective in moving air through the auditorium as the fan used in primarily used as a mixing fan and cannot generate high differential pressures; there are also large gaps between the fan blades and the interior surfaces of the stacks. The rises in temperatures due to the fan are visible in figures 4.9 and 4.10.

The variability of air velocities in the stacks is evident from the temperature fluctuations of the stack air (see figure 4.11). The internal air temperature remains almost constant during the same period. The asymmetric nature of air flow is indicated in figure 4.12, where air speeds on the right hand side (facing the front of the theatre) are considerably greater than those on the left hand side

The following conditions were recorded for "summer" external conditions (i.e. average external air temperatures at or above approximately 15 °C):

#### 4.7.4(a) Inlets and outlets closed

1) Room air temperatures were generally in the range 21 to 23 °C even when high external temperatures occurred (i.e. above 27 °C) (see figures 4.9 and 4.10).
2) The spatial variation in ceiling slab temperature was about 4 °C (max. 24 °C to 27 °C).
3) Low air velocities: Generally inlet and room air velocities were low (less than 0.3 m/s on average).

#### 4.7.4(b) Inlets and outlets open

1) Room air temperatures were generally in the range 22 to 26 °C even when high external temperatures occurred (i.e. above 27 °C).
2) Ceiling temperatures were slightly higher than in mid-season, but behaved in a similar fashion.
3) Inlet air temperatures were the same or slightly less than external temperatures (temperatures could be depressed due to heat transfer to the precooled underfloor slap).
4) Corresponding air velocities could be high (up to 0.5 m/s); however this was not likely to cause discomfort due to the higher temperatures involved (i.e. 18 to 34 °C).
5) Air speeds on the right hand side were generally considerably higher than those on the left hand side (due to the asymmetric layout of the inlet openings in the external wall) (see figure 4.12).
6) High levels of turbulence were encountered, apparently generated by rapidly fluctuating wind induced pressures.

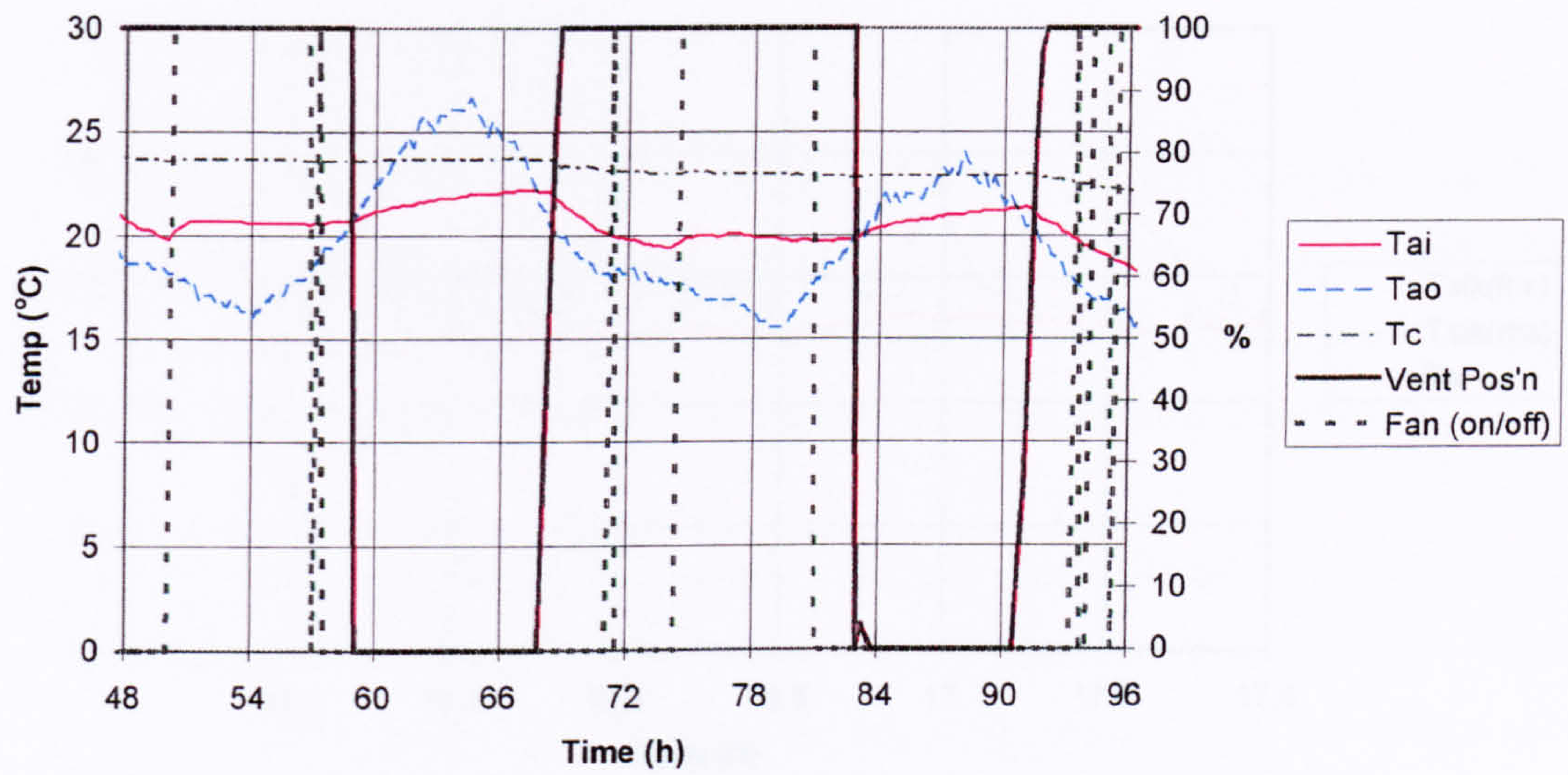



Figure 4.10 Internal, ceiling slab and external temperatures (with night venting)

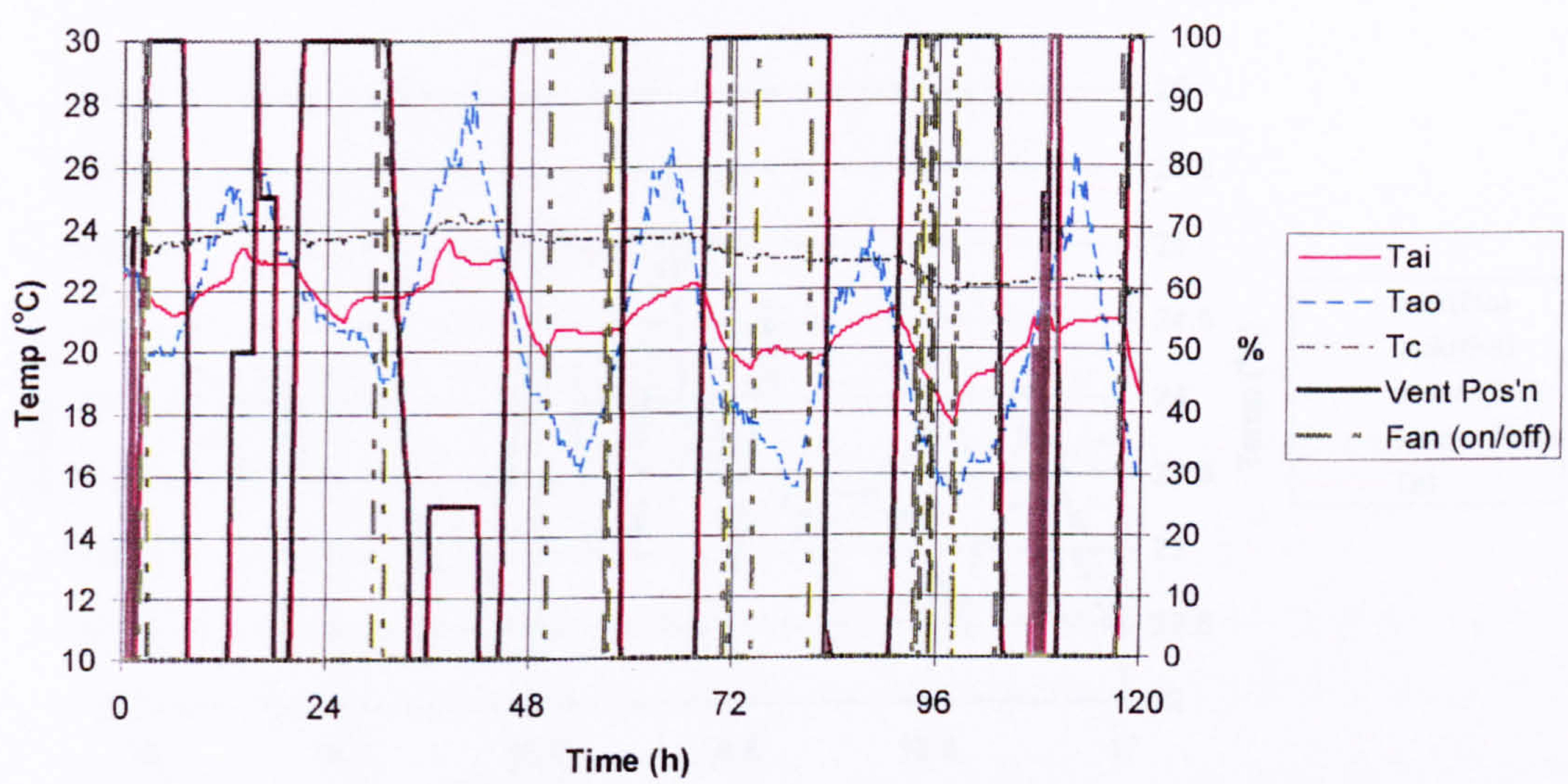



Figure 4.11 Internal, ceiling slab and external temperatures (with night venting)

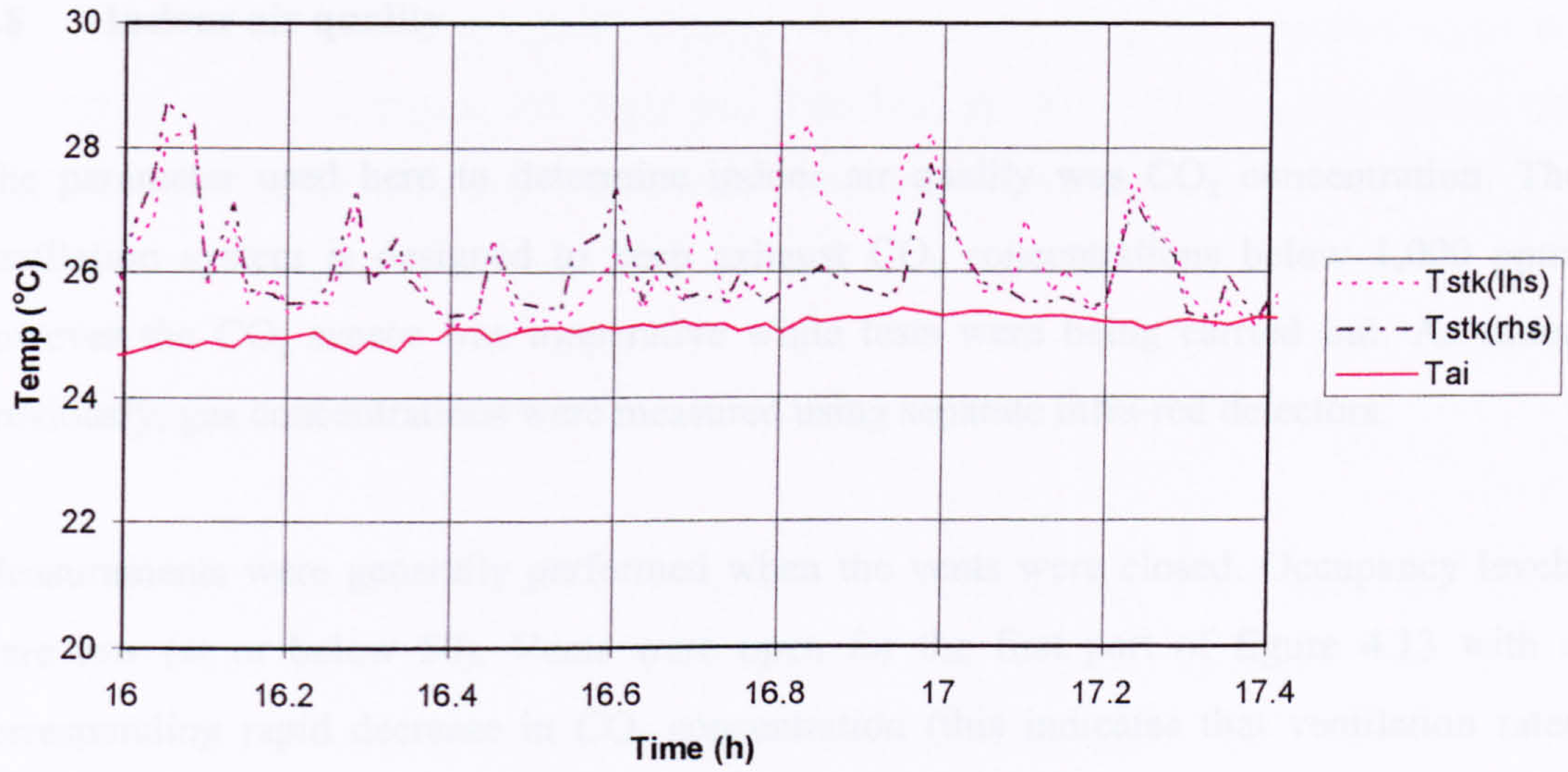



Figure 4.12 Internal and stack air temperatures for summer external conditions

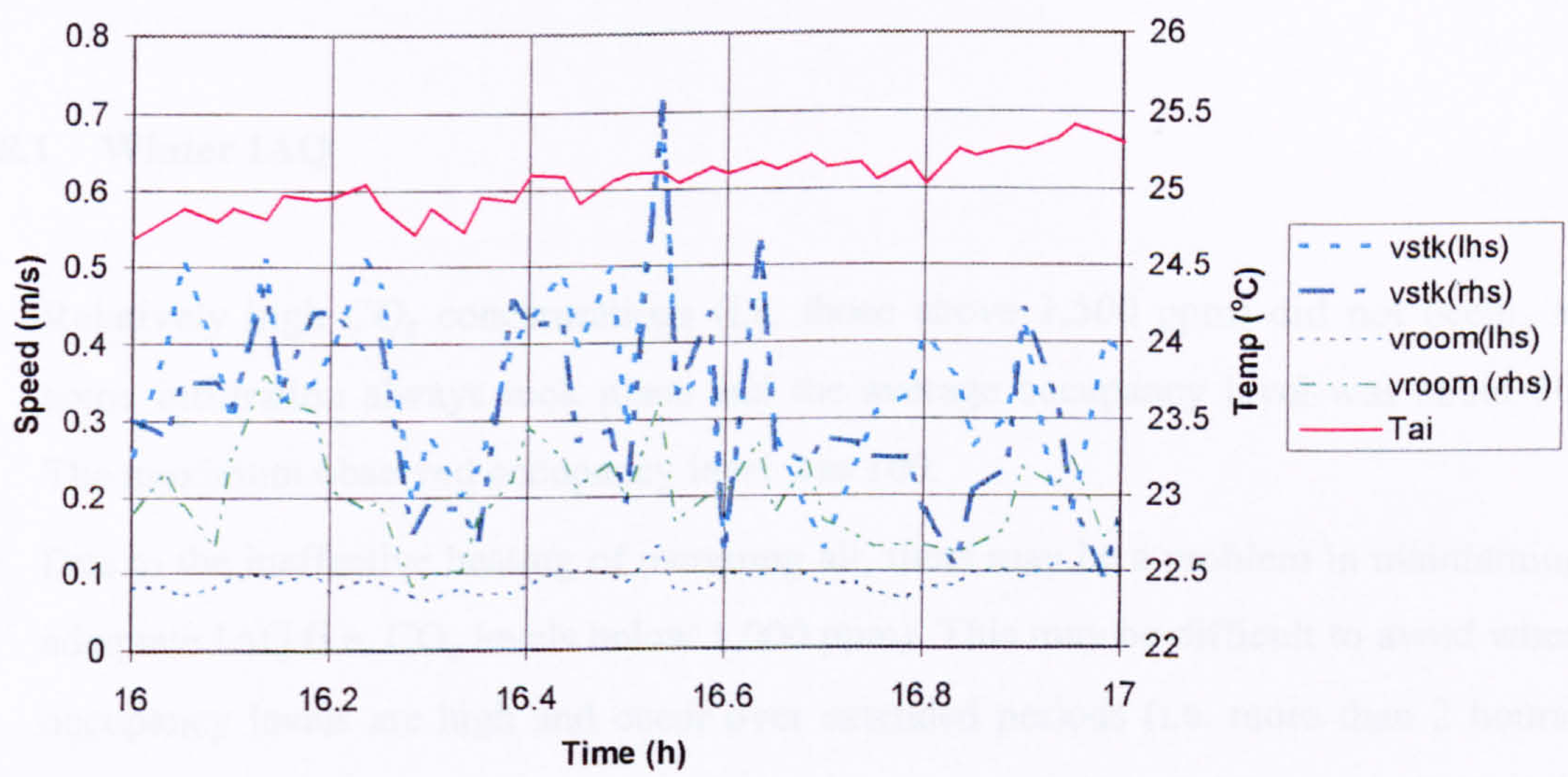



Figure 4.13 Room and stack air speeds for summer external conditions

## 4.8 Indoor air quality

The parameter used here to determine indoor air quality was $CO_2$ concentration. The ventilation system is designed to keep exhaust $CO_2$ concentrations below 1,000 ppm, however the $CO_2$ sensor was inoperative while tests were being carried out. As stated previously, gas concentrations were measured using separate infra-red detectors.

Measurements were generally performed when the vents were closed. Occupancy levels were low (at or below 50). Vents were open for the first part of figure 4.13 with a corresponding rapid decrease in $CO_2$ concentration (this indicates that ventilation rates were high, i.e. greater than 10 air-changes per hour). There-after rises in gas concentrations are due to the presence of people and drops due to their absence. Figure 4.14 exhibits similar information, and the outside concentration was also measured; this remained almost constant at about 300 ppm. The maximum concentration measured was 1,300 ppm.

### 4.8.1 Winter IAQ

1) Relatively high $CO_2$ concentrations (i.e. those above 1,500 ppm) did not occur, as some infiltration always took place and the average occupancy level was about 50. The maximum observed occupancy level was 100.
2) Due to the ineffective heating of incoming air, there may be a problem in maintaining adequate IAQ (i.e. $CO_2$ levels below 1,000 ppm). This may be difficult to avoid when occupancy levels are high and occur over extended periods (i.e. more than 2 hours) and low external air temperatures (i.e. temperatures below 5 °C) are indicated if discomfort is to be avoided.
3) Higher local air change rates were recorded on the right hand side of the room than on the left hand side (see figure 4.22).

4) $CO_2$ concentrations correlated strongly with average room temperatures when the vents were closed (i.e. the emissions and heat gains from occupants caused the simultaneous increase in $CO_2$ levels and room air temperatures).

### 4.8.2 Mid-season IAQ

1) Maximum measured $CO_2$ concentration was approximately 1,400 ppm (see figure 4.14). Thus levels were well below the upper "safe" limit of 5,000 ppm but slightly above the recommended maximum concentration of 1,000 ppm. Thus conditions were deemed to be acceptable. Rapid falls in concentration occurred (down to the external level of c. 300 ppm) when the vents were partially opened.
2) Higher local air change rates were recorded on the right hand side of the room than on the left hand side (see figure 4.23).
3) Point number 4) above (section **4.8.1**) is also applicable.

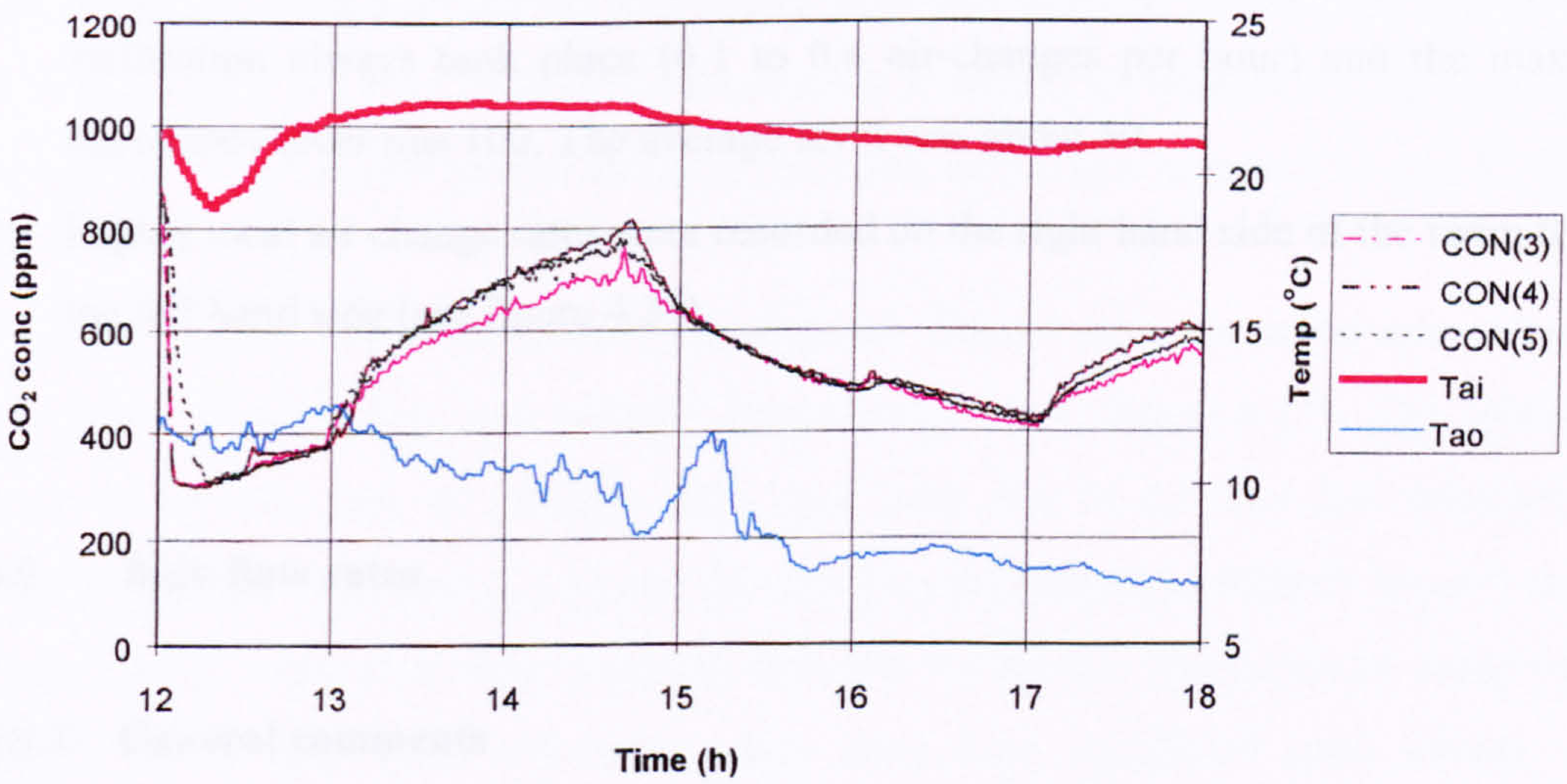



Figure 4.14 $CO_2$ concentrations (on the lhs and rhs and at high level, lhs), internal and external air temperatures versus time (maximum occupancy level equalled 50). Infiltration only.

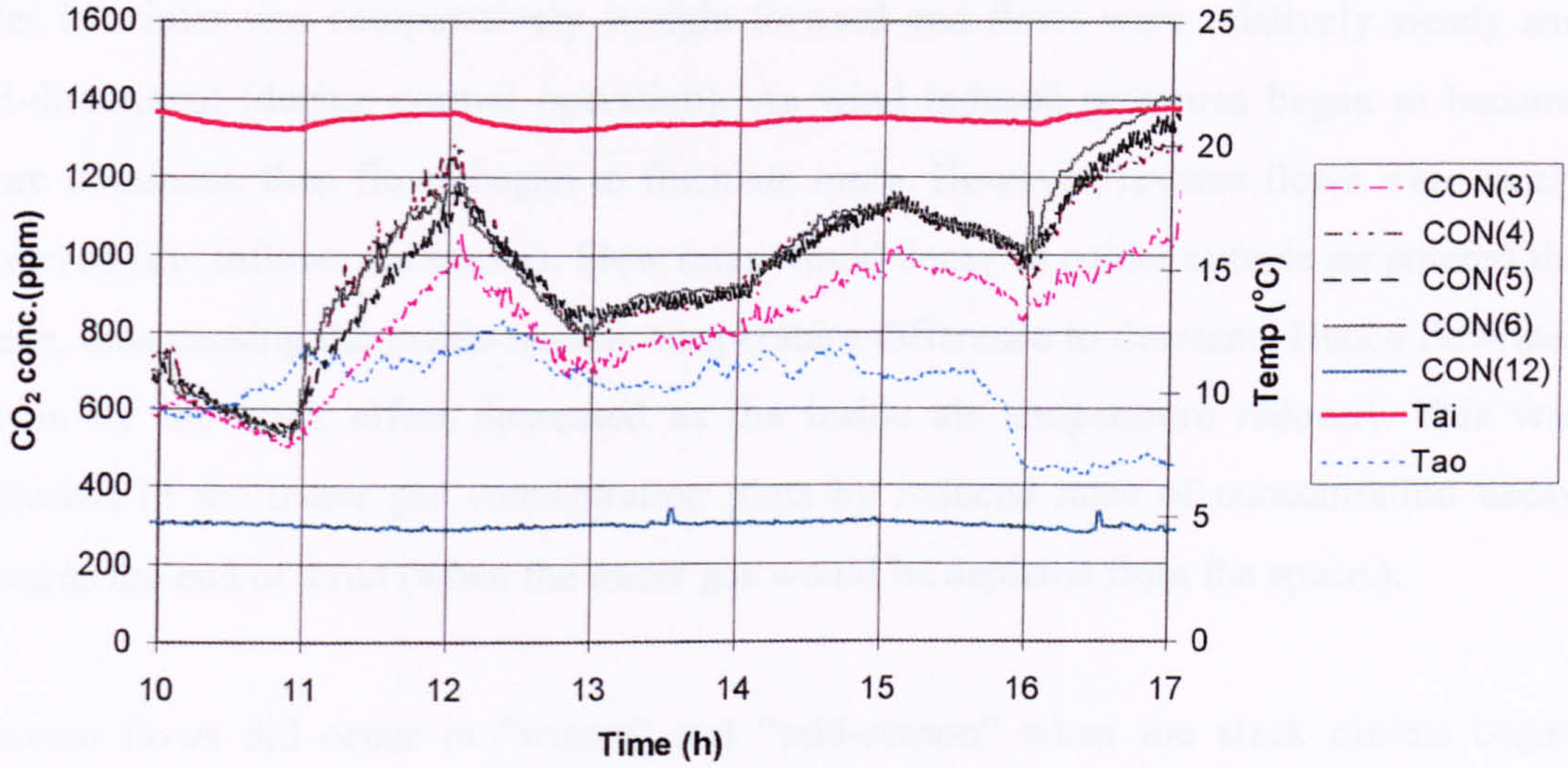



Figure 4.15 $CO_2$ concentrations (in the stacks, on the lhs and rhs and at high level, lhs, rhs and outside) and internal air temperature versus time (maximum occupancy level equals 50). Infiltration only.

### 4.8.3 Summer

1) Excessive $CO_2$ concentrations did not occur (for average occupancy levels). Some infiltration always took place (0.1 to 0.4 air-changes per hour) and the maximum occupancy level was 100. The average level was about 50.
2) Higher local air change rates were recorded on the right hand side of the room than on the left hand side (see figure 4.24).

## 4.9 Bulk flow rates

### 4.9.1 General comments

Flow rates are driven largely by buoyancy forces in winter and by a combination of buoyancy and wind induced forces in mid-season and summer. The measurement of flow rates in winter was comparatively straight-forward and flows were relatively steady and uni-directional (during normal operation). As wind induced pressures began to become more dominant, then flows began to fluctuate more. However, reverse flows were rarely observed (i.e. inflows via stacks). Flow rates would decay as colder outside air entered the space, thus causing the inside-outside temperature difference to decrease. Hence flowrates driven by the stack effect decreased as the inside air temperature reduced. This was indicated in the tracer gas concentration plots by reduced rates of concentration decay towards the end of a run (when the tracer gas would be depleted from the space.).

Reverse flows did occur in "winter" and "mid-season" when the stack outlets began moving to their closed positions. Stack temperatures provided useful information on flow directions through the stacks; if there was a significant difference between inside and outside temperatures, then a sharp drop in temperature would indicate that air was flowing down a chimney while a rapid rise in temperature would demonstrate that up-flows were occurring. In addition, individual velocity values within each stack were approximately the same over a given time period (see figures 4.15 and 4.16).

There was no evidence that two way flows were taking place within the relatively large stacks (height c. 18.5 m; internal cross-sectional width 3.5 m; depth 1.43 m); temperatures measured using velocity probes indicated the same values (see figures 4.15 and 4.16).

There was some correlation between average air change rates measured using tracer gas and those found using low velocity anemometers (see figure 4.17). The differences between the two sets of readings may have been due to the fact that relatively few sampling points were used (i.e. 6) and the sampling interval was relatively large (1 minute) relative to the ventilation time constants detected. Ventilation measurement using velocity probes would only work effectively when there were significant stack effects, which would produce large relatively constant flowrates.

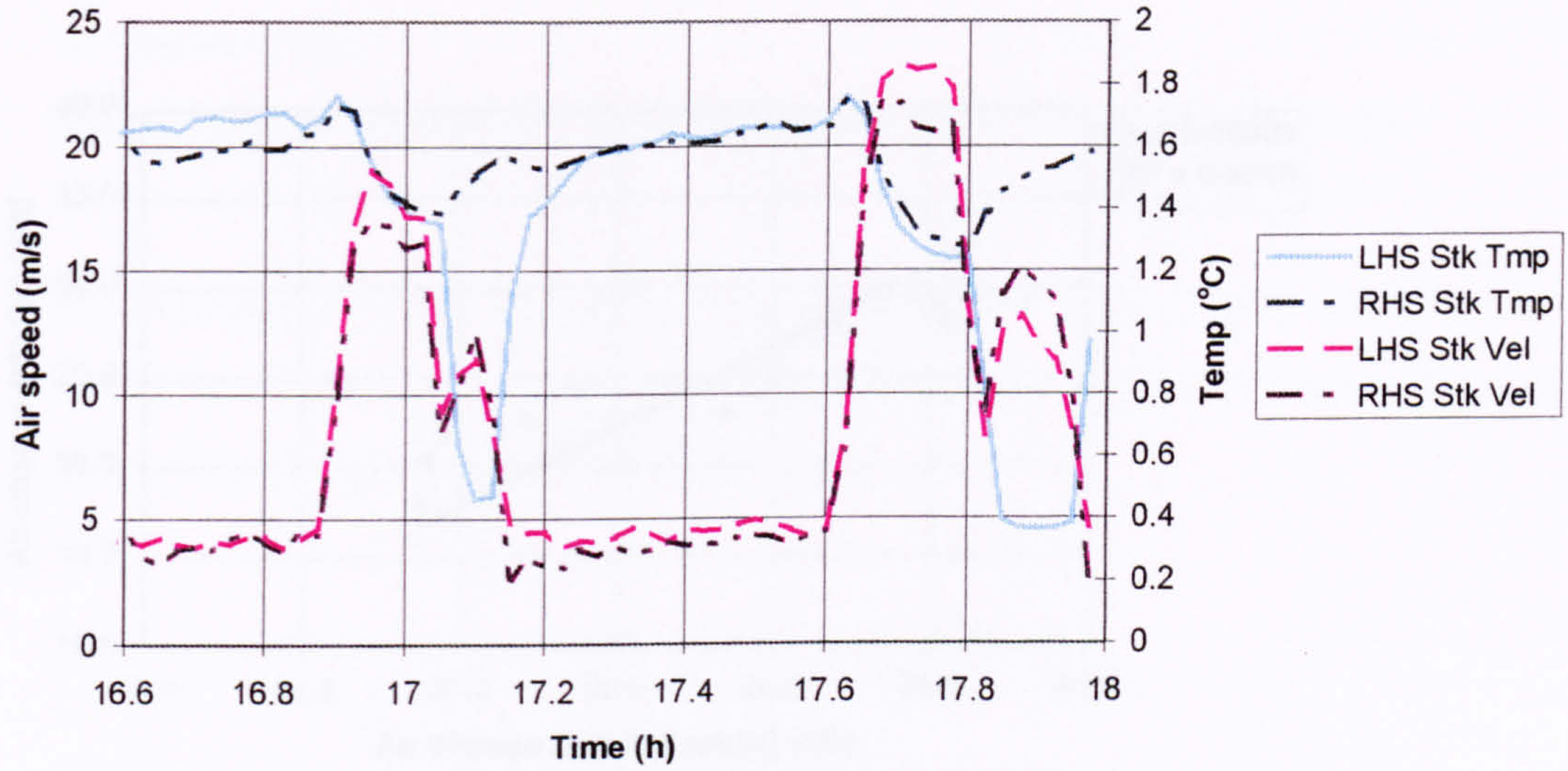



Figure 4.16 Stack temperatures and air velocities (winter)

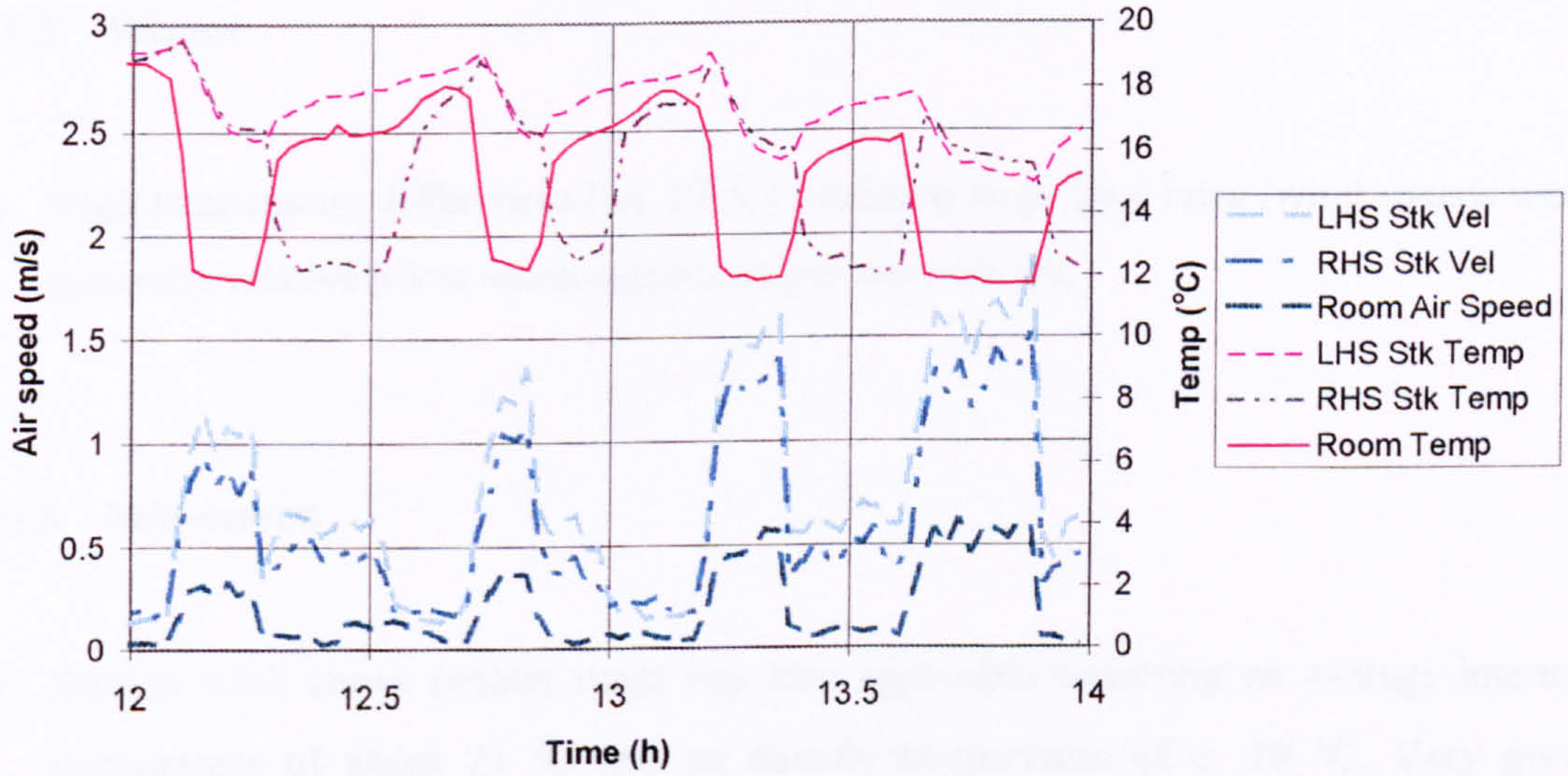



Figure 4.17 Stack, room air velocities and temperatures; (mid-season)

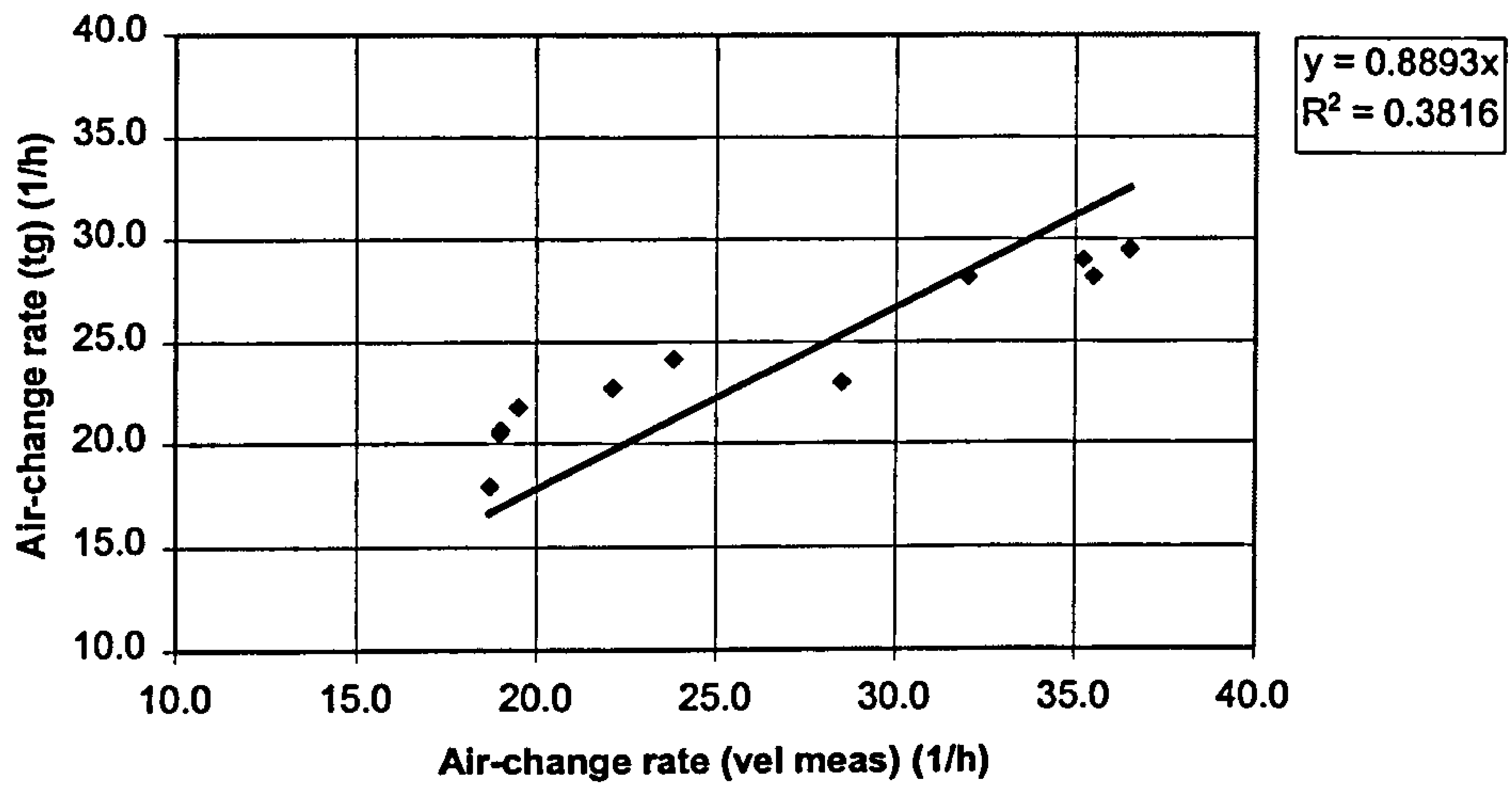



Figure 4.18 Comparison of air change rates found using the tracer gas decay method and from low velocity anemometry recordings.

### 4.9.2 Winter

1) High temperature differences (i.e. 15 °C) produced large flow rates (wind speeds were generally relatively low when measurements were taken).

### 4.9.3 Mid-season

1) Section 4.9.2 above (winter runs) was also applicable assuming an average internal temperature of about 21 °C and an outside temperature of c. 10 °C. Very good agreement was reached between flow rates measured using stack mounted velocity probes and values calculated using the CIBSE Guide stack formula [18] , i.e.

$$q_{vB} = C_d A_B \left[ 2gh_a \left( \frac{T_i - T_o}{\frac{T_i + T_o}{2}} \right) \right]^{0.5} \qquad (4.1)$$

(see figure 4.18)

(assuming wind speeds were "low" (less than 2.7 m/s at stack exhaust level)

Some air-change rates were greater than that predicted by the CIBSE formula; this is due to the effects of wind causing greater differential pressures between inlets and outlets (than that produced by buoyancy only).

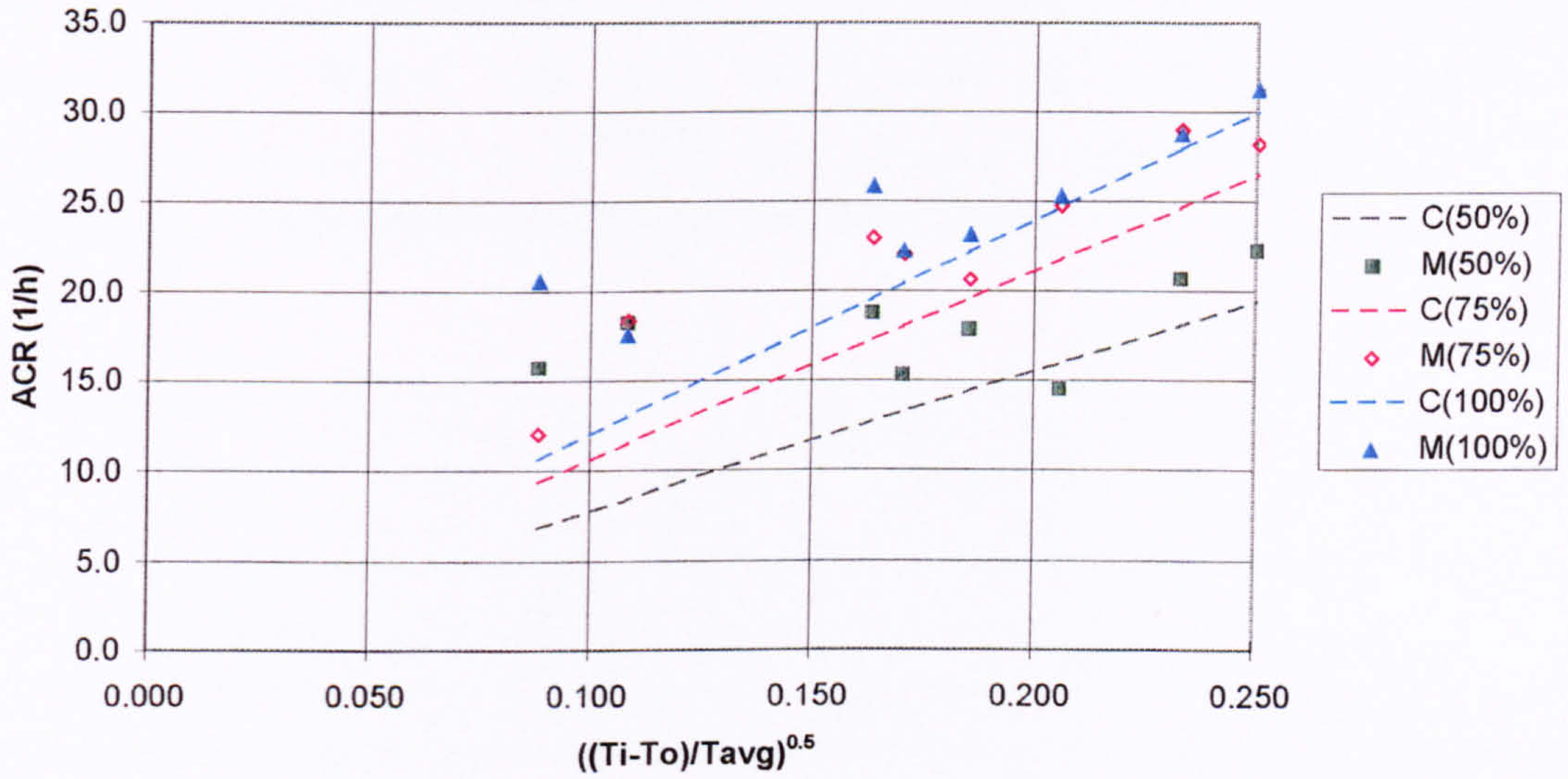



Figure 4.19 Airchange rate ($h^{-1}$) versus $((T_{ai}-T_{ao})/T_{avg})^{0.5}$ for three different opening settings
(C = calculated values; M = measured values)

### 4.9.4 Summer

1) Flow rates were driven by stack and wind effects. Some instances occurred when the internal air temperature matched the external air temperature. Flow rates were generated even when these conditions took place, suggesting that wind effects predominate. Upflows in the stacks were recorded even when a zero or negative temperature differential prevailed.

2) Flow rates generated at night (i.e. night venting) appeared to reduce room surface temperatures (see figure 4.19). Night venting appears to partly prevent surface temperatures rising. Figure 4.20 indicates that the surface temperatures gradually increase if night venting is not carried out.

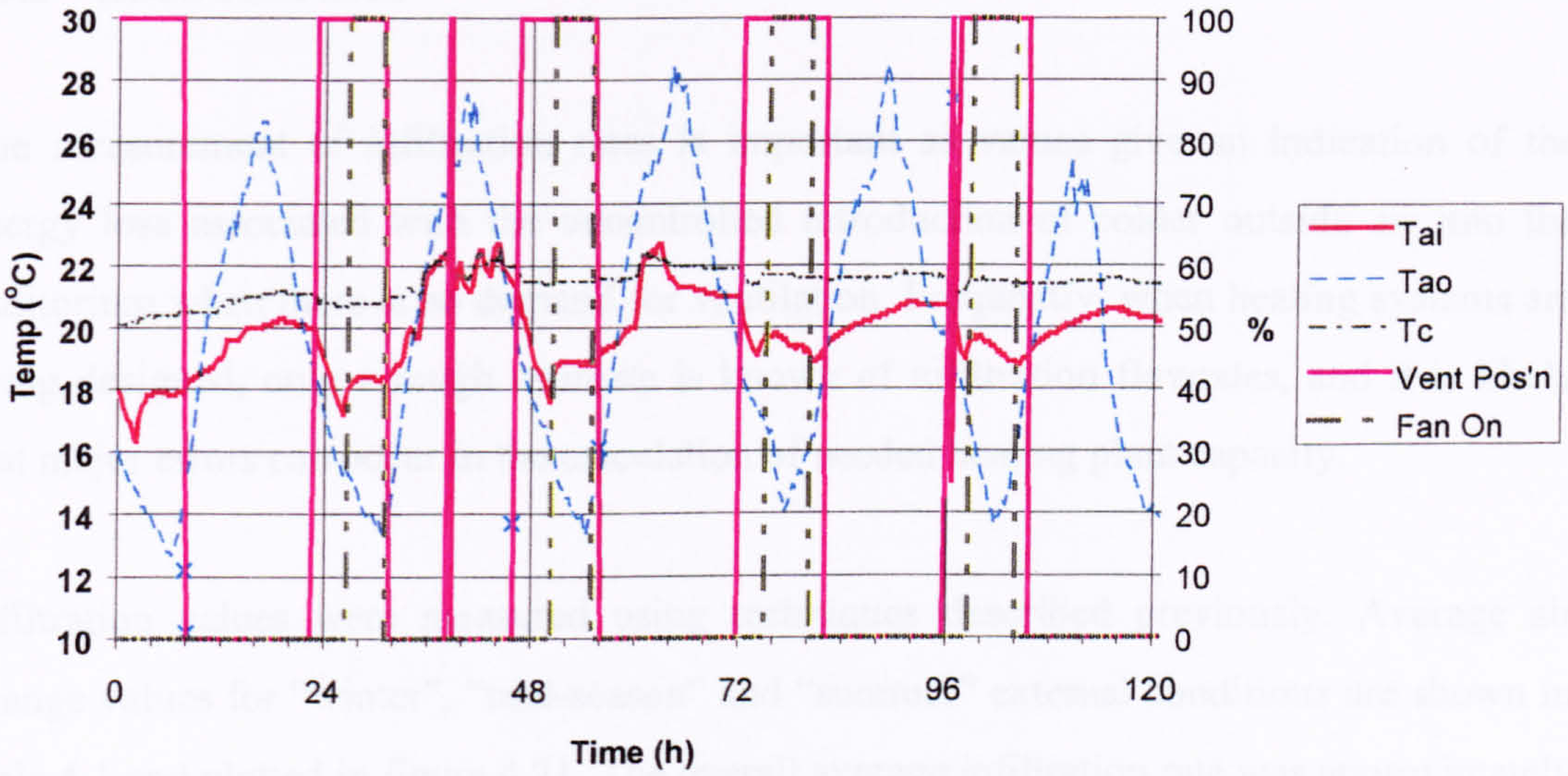



Figure 4.20 Summertime temperatures with night venting

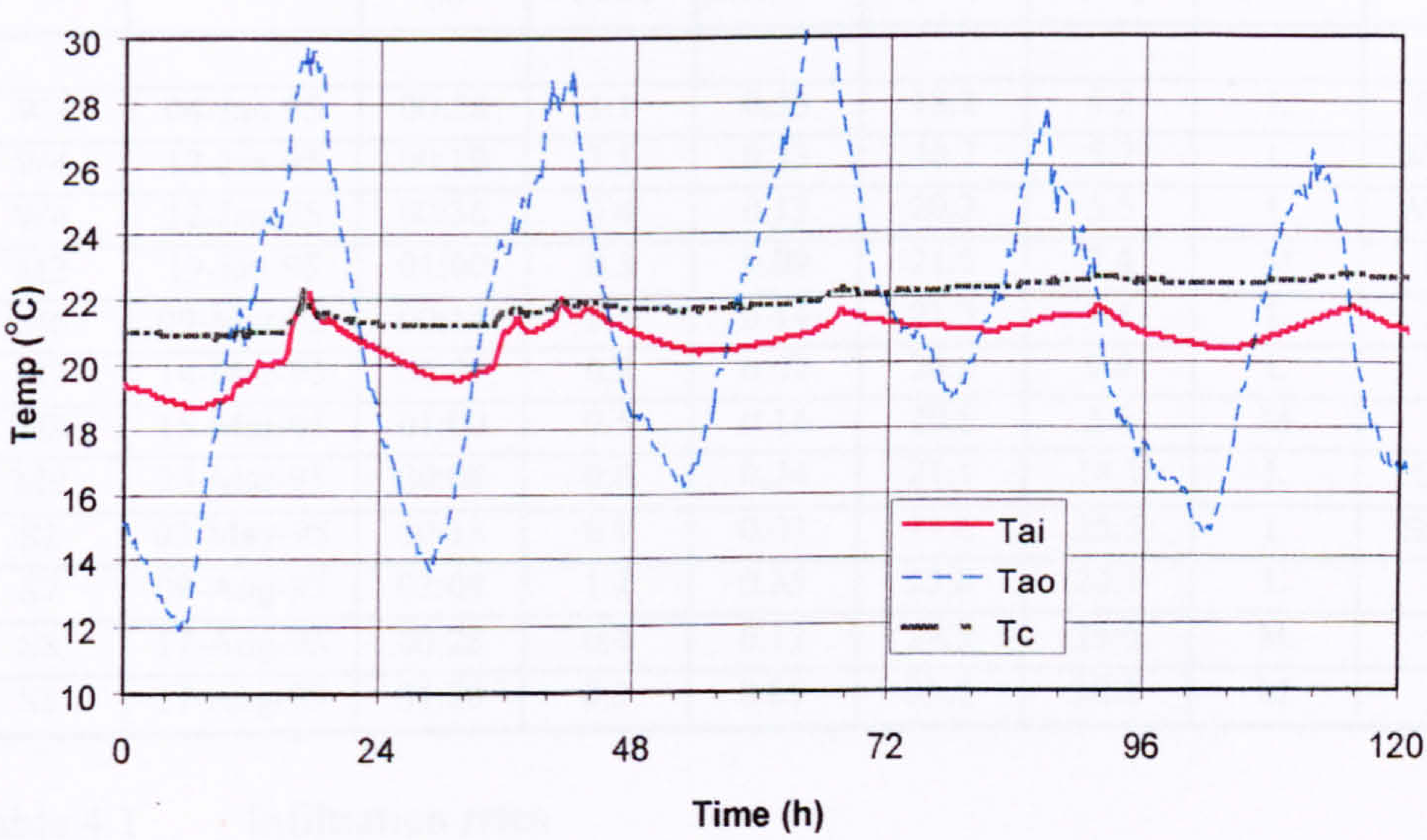



Figure 4.21 Summertime temperatures without night venting

### 4.9.5 Infiltration rates

The measurement of infiltration rates is important as values give an indication of the energy loss associated with the uncontrolled introduction of colder outside air into the auditorium when there is no demand for ventilation. Frequently, when heating systems are being designed, only a rough estimate is known of infiltration flowrates, and it is likely that major errors can occur in the calculation of needed heating plant capacity.

Infiltration values were measured using techniques described previously. Average air change values for "winter", "mid-season" and "summer" external conditions are shown in table 4.1 and plotted in figure 4.21. The overall average infiltration rate was approximately 0.6 ($h^{-1}$) or about 0.2 $m^3/s$.

| Run | Date | Time Duration (h) | Avg ACR (1/h) | Avg Flowrate (m3/s) | Avg. Temp. Internal (°C) | Avg. Temp. External (°C) | Wind Speed (m/s) | Wind Dir'n |
|---|---|---|---|---|---|---|---|---|
| | | | | | | | | |
| W3 | 04-Jan-95 | 00:58 | 1.1 | 0.33 | 18.2 | 5.2 | L | SSE |
| W4 | 12-Jan-95 | 00:19 | 1.1 | 0.33 | 20.7 | 5.7 | L | WNW |
| W4 | 12-Jan-95 | 00:36 | 0.4 | 0.12 | 20.3 | 5.5 | L | WNW |
| M2 | 19-Jan-95 | 01:00 | 0.3 | 0.09 | 21.5 | 7.4 | M | SE |
| M6 | 09-Mar-95 | 00:14 | 0.5 | 0.14 | 21.3 | 9.1 | L | S |
| M7 | 14-Mar-95 | 00:59 | 0.2 | 0.07 | 20.9 | 9.9 | L | W |
| M8 | 15-Mar-95 | 01:00 | 0.5 | 0.14 | 20.5 | 3.6 | M | W |
| M9 | 23-Mar-95 | 00:08 | 0.8 | 0.24 | 21.1 | 14.1 | L | SW |
| S2 | 02-May-95 | 00:18 | 0.1 | 0.03 | 21.6 | 25.5 | L | SSW |
| S7 | 09-Aug-95 | 02:08 | 1.2 | 0.35 | 23.8 | 23.1 | L | E |
| S8 | 17-Aug-95 | 00:28 | 0.4 | 0.12 | 24.3 | 25.6 | M | E |
| S8 | 17-Aug-95 | 01:20 | 0.3 | 0.09 | 25.8 | 28.5 | M | E |

Table 4.1 Infiltration rates

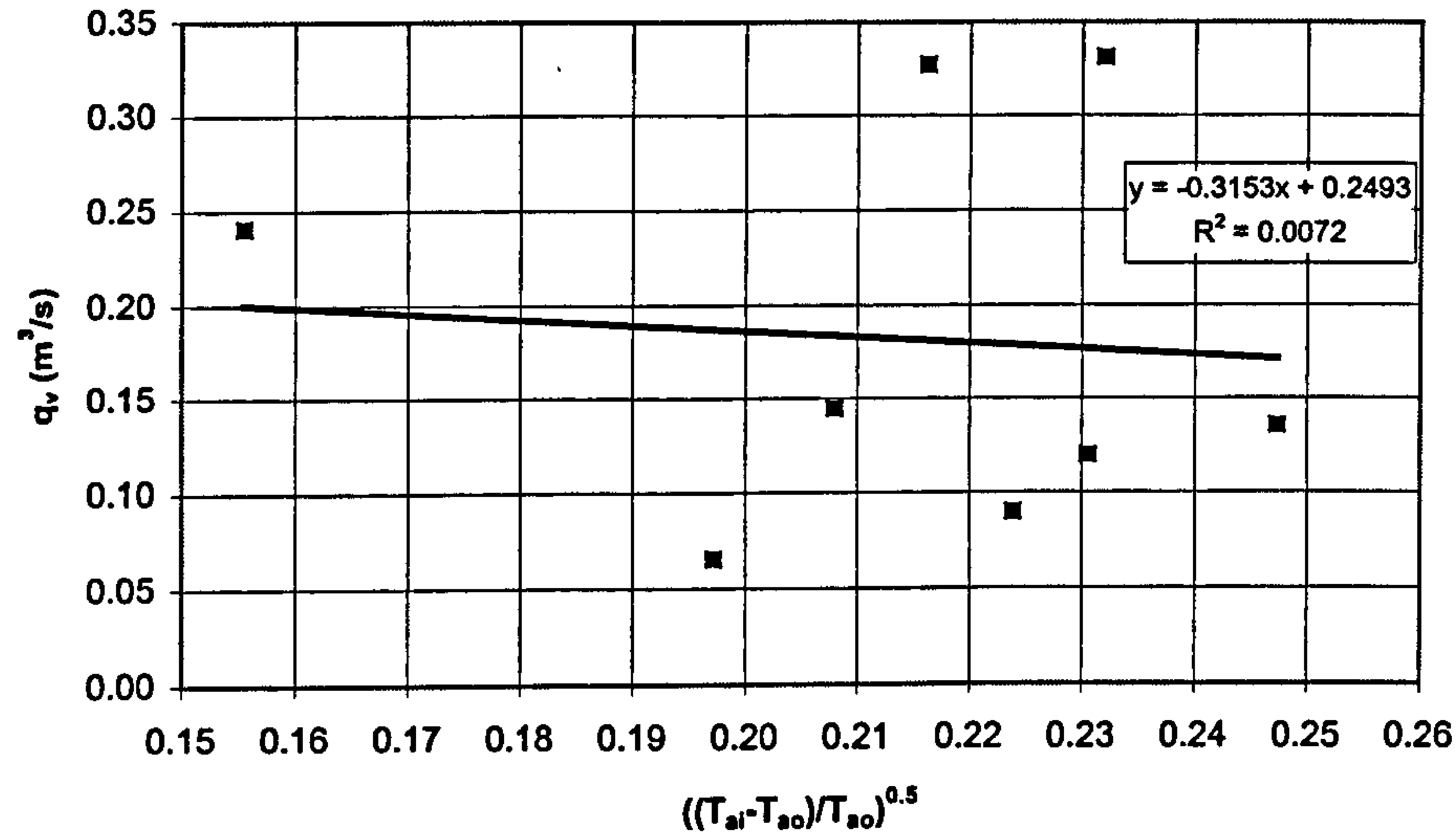



Figure 4.22 Infiltration versus $[(T_{ai}-T_{ao})/T_{ao}]^{0.5}$

It can be seen from figure 4.21 that no discernible correlation occurs between a function of inside and outside temperatures and infiltration flowrates. Possible explanations are that the lowest effective area settings (those when the space is unoccupied) cannot be consistently reproduced every time the inlet dampers and exhaust openings close or return to their default settings, and that there are additional leakage paths (e.g. through gaps around the external openings, through internal doorways or through window frames).

Infiltration rates are probably driven mainly by wind effects as there appears to be no co-relation between a function of inside-outside temperature differences and flowrates. An average infiltration rate of 0.6 air-changes per hour was calculated. This corresponds roughly to a leakage rate of 11 $m^3\ h^{-1}\ m^{-2}$ (of external area).

## 4.10 Air distribution and air change indices.

Air change rates were found at six locations (see figure 3.4) within the space for winter, mid-season and summer external conditions. The monitored zones were (i) the lhs stacks, (ii) the rhs stack, (iii) the lhs (low level), (iv) the rhs (low level), (v) the lhs (high level) and (vi) the rhs (high level).

Air change indices were calculated by dividing the air change rate in a particular zone by the average air-change rate over the 6 zones (see section 3.5.1, p.54 for the equations used). These give a useful indication as to the effectiveness of the ventilation system; a value below 1 is deemed to be unsatisfactory.

There does not appear to be much variation in air change index value when opening areas and hence flow rates are increased. The actual values are broadly similar (i.e. from 0.8 to 1.0) for each zone except the rhs zone (the zone facing inlet plenums) where much higher air speeds were experienced. Indices in this region ranged from 1.2 to 1.7. It is clear if the average air change values are examined that the system has no difficulty in providing relatively large volumes of air but, as stated previously, high levels of occupant discomfort are likely in winter and mid-season when opening settings are greater than or equal to 50 %. The degree of air flow control is also relatively limited.

The indices for different opening settings are presented in figures 4.22, 4.23 & 4.24.

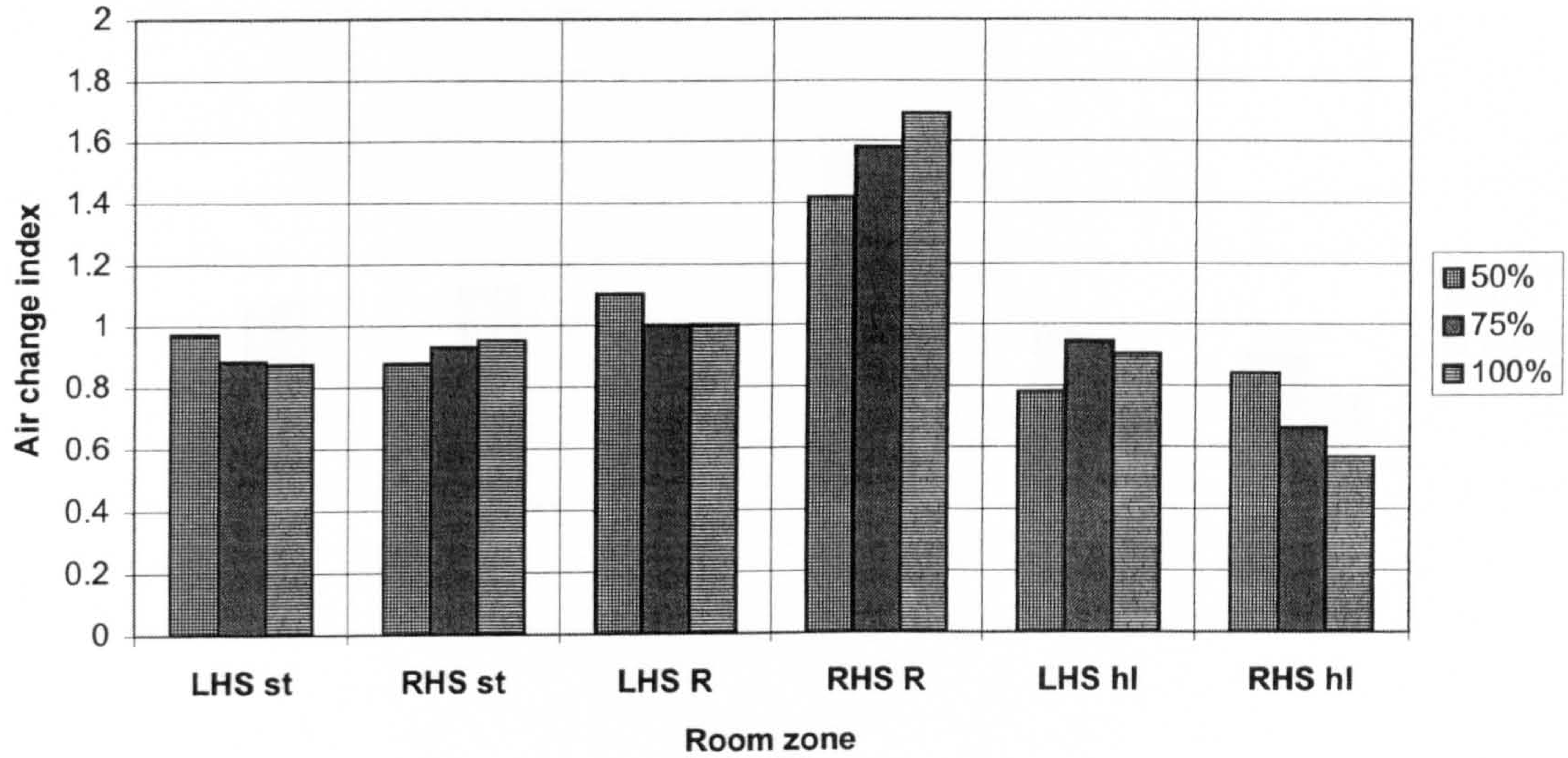



Figure 4.23. Average air change indices in winter for 3 different opening settings (BMS damper settings of 50 %; 75 % and 100%; LHS st = left hand side stack; RHS st = right hand side stack; LHS R = left hand side of room; RHS R = right hand side of room; LHS hl = left hand side high level; RHS hl = Right hand side high level)

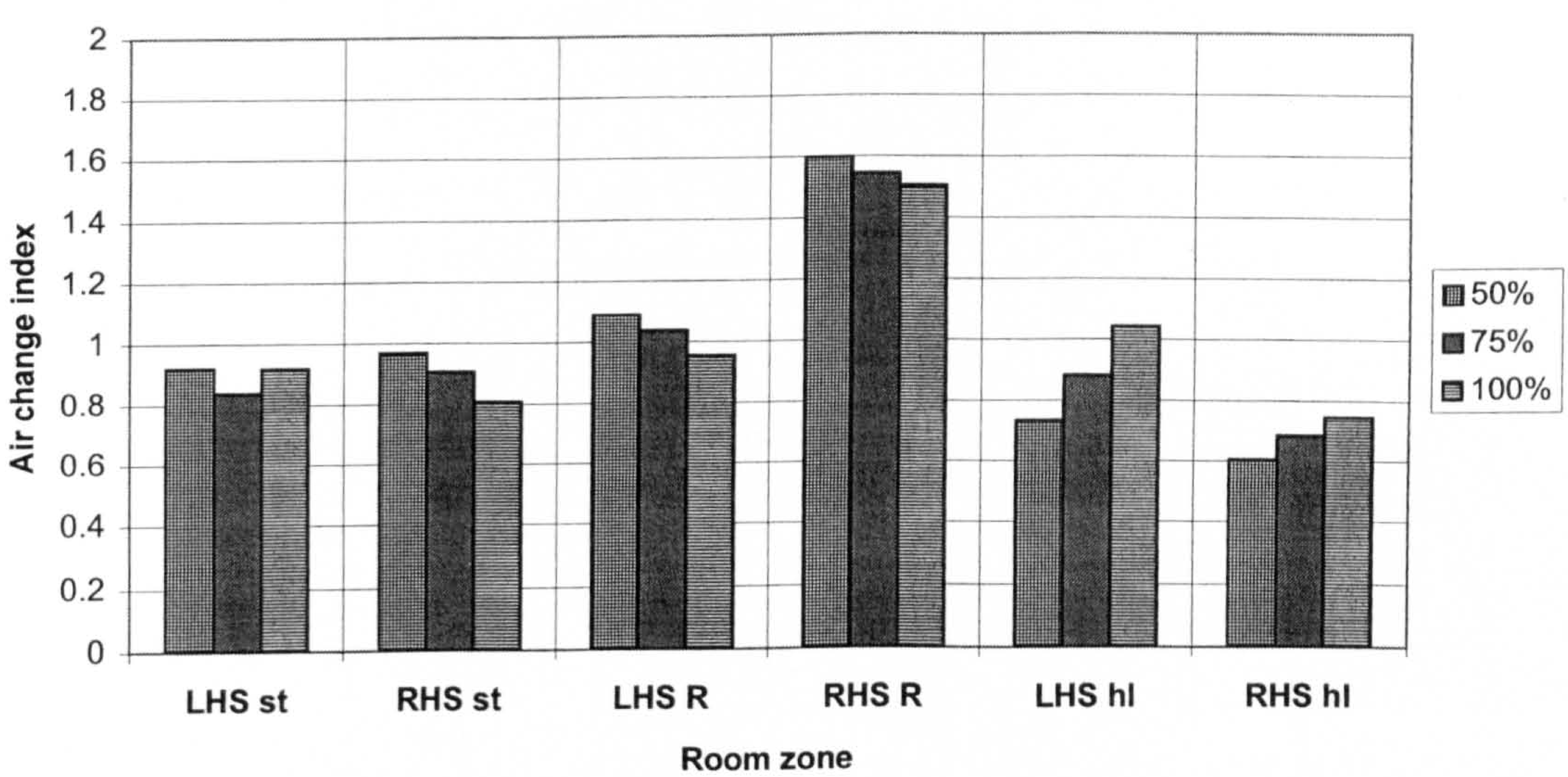



Figure 4.24 Average air change indices in mid-season for 3 different opening settings

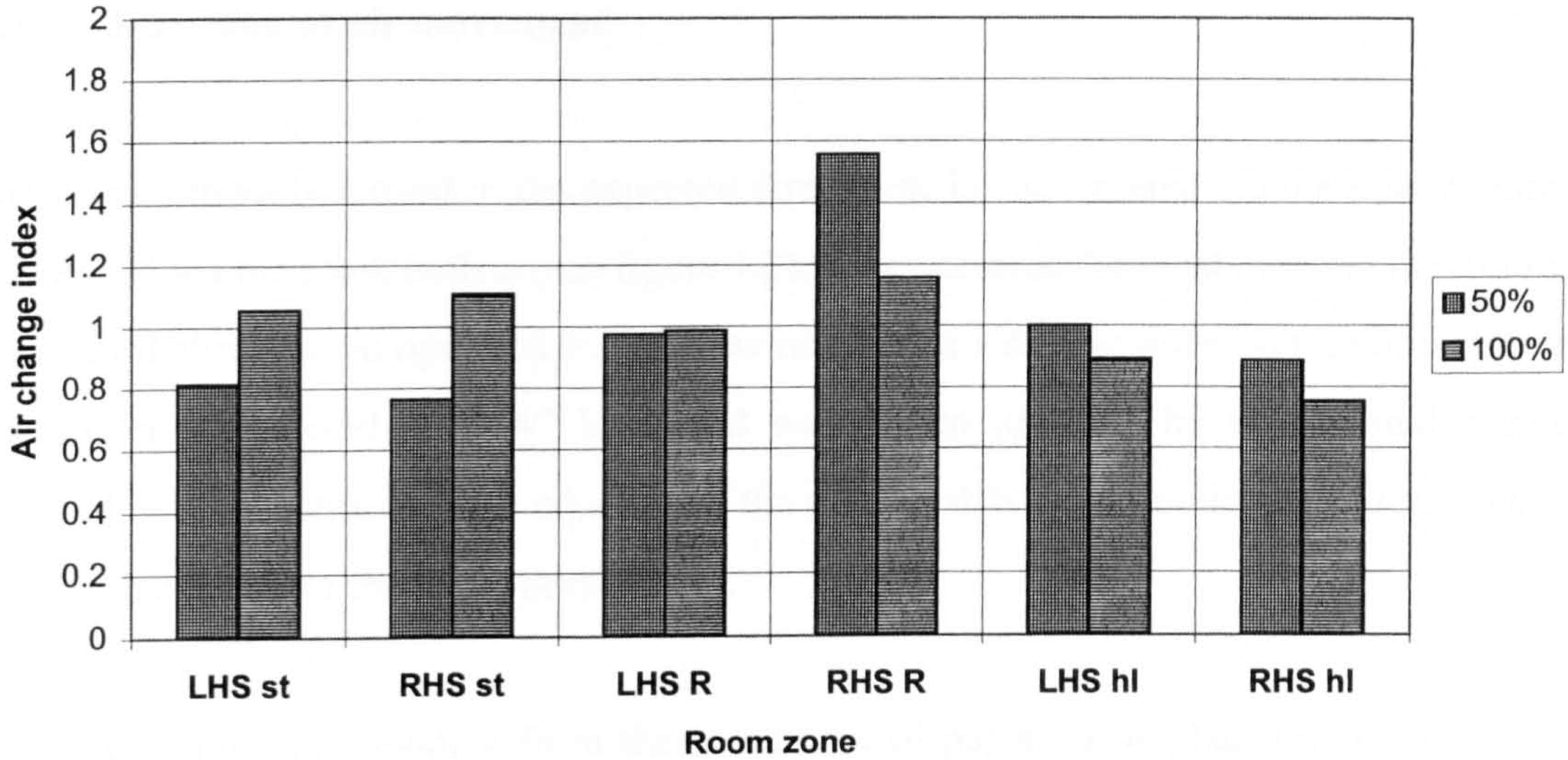



Figure 4.25 Average air change indices in summer for 2 different opening settings

## 4.11 Direction of air movement

Air flows generally moved in the expected directions, i.e. air entered via the inlet plenums and exited via the stack outlets (see figure 1.3). This occurred for most climatic conditions. The ventilation system operated in a similar manner to a displacement ventilation system; i.e. colder air entered at "low" level and was driven upwards by the thermal plumes induced by occupants. A zone adjacent to the ceiling slab would collect warmer, stale air and discharge it into the two stacks.

Flow directions were evident from the movement of paper strips placed across the stack openings, from the behaviour of smoke and from recorded stack temperatures (see figure 4.16).

Reverse flow occurred (as stated above) when stack vents commenced closing. In this situation, one stack would temporarily present a higher resistance to flow than the other stack and flow would be take the path of least resistance, i.e. it would exit via the stack presenting the lowest resistance to flow. Hence flow from outside would be sucked down one chimney and discharged out the second chimney. No air could enter via the inlet louvres, since these shut within 2 minutes while the outlets take a total of 8 minutes to close. The phenomenon is illustrated in figure 4.25.

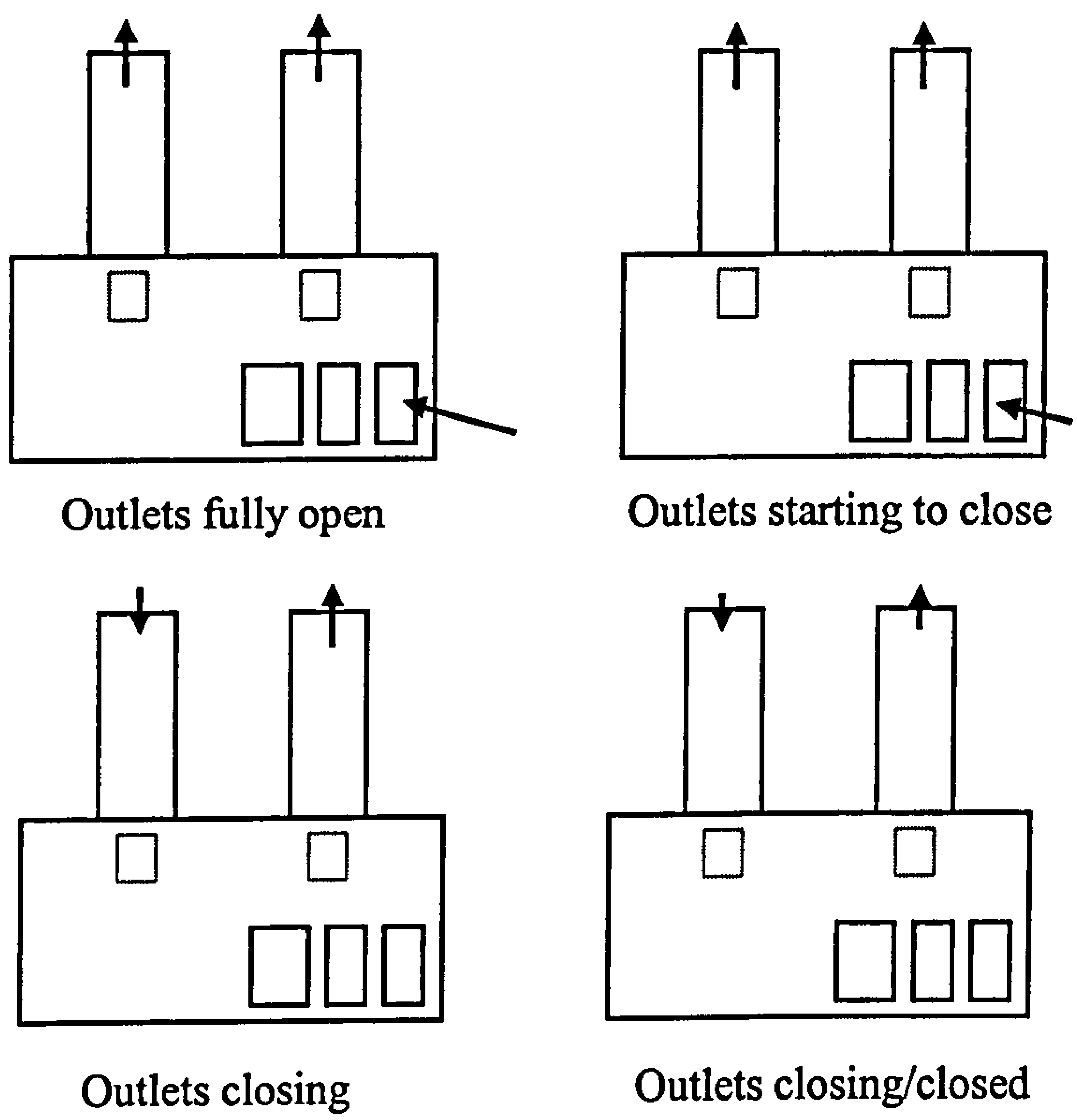



Figure 4.26 Generation of downflows in the lhs stack

## 4.12 Conditions found in lecture theatre 1.12

### 4.12.1 General comments

Detailed measurements of temperature, air speeds and local air change rates have not been carried out in lecture theatre 1.12 (plan and section are shown in figure 1.4). However, a number of spot measurements were performed and qualitative results of conditions in this space have been found.

### 4.12.2 Comfort parameters

In winter, relatively large variations in air temperature with height above the lowest floor level were noted (i.e. 2 to 4 °C) when the heating was operating and with vents in their closed positions. Thus temperature stratification was taking place. This effect also occurred in mid-season and to a lesser extent in summer (no heating). Air speeds adjacent to inlet grilles (at ankle height) were very low (less than 0.1 m/s). However, significant air speeds did occur at the locations of the large inlet openings at the front of the auditorium (c. 0.8 m/s) which could cause discomfort in winter, even with the heating switched on.

### 4.12.3 Bulk flow rates

Air circulation (from the front inlets to the stack outlets) was as shown in figure 1.4 and large air speeds were found at the faces of the two large inlets (between finned heating tubes). Air then moved towards the stack openings and out of the stack outlets (with little or no air flow rates taking place through the floor plenum and via the room inlet grilles, as was intended by the design engineers). Hence the distribution of fresh air was likely to be relatively non-uniform with most of the air being received by occupants sitting adjacent to the stepped central isle.

It was easy to demonstrate that significant bulk air movements were taking place in the theatre. If the inlets and outlets were opened in winter or mid-season, then air speeds were sufficiently high to drive the extract fan mounted in the right hand side stack (i.e. velocities above c. 0.5 m/s). This was an attractive way of demonstrating the effectiveness of the natural ventilation system used to visitors of the building.

### 4.12.4 Indoor air quality

For the reason outlined above (i.e. relatively poor fresh air distribution), the average IAQ is likely to be inferior to that experienced in theatre 1.10. However, due to intermittent nature of occupancy, average $CO_2$ levels are likely to be of the same order of magnitude (i.e. up to 1,400 ppm for an occupancy level of 75).

## 4.13 Examination of the effects of wind induced pressure differences on flow rates

### 4.13.1 Introduction

A series of tests were performed from July to August 1995 to examine the effects of changing the outlet configurations on flow rates through the auditorium. It had been determined that wind induced pressures had significant effects on flow rates when buoyancy effects were negligible or if the outside air temperature was several degrees higher than the inside temperature (i.e. during the summer). If this was the case, then altering the outlet opening settings should affect flow rates. In theory, higher flow rates ought to occur when windward facing openings are closed and leeward facing openings are open. [19]

In general, at least three modes of stack operation can be envisaged when wind induced pressures are higher than those produced by buoyancy. The first is as described above, i.e. when windward vents are closed (being positively pressurised) and leeward vents are open (being negatively pressurised). The second is the opposite of this; this would occur when windward vents are open and leeward vents are closed. The third would occur when both windward and leeward vents are open and a venturi effect would cause a suction to occur at the top of a chimney thus causing outflows from the space. In practise, this configuration would only occur by the deliberate use of the manual controls, the normal operation of the outlets causes all openings to be operated "simultaneously". However, a complication is that a period of up to 8 minutes is required for the rack and pinion opening mechanisms to work, i.e. each actuator motor operates in series to avoid causing excessive starting currents which could cause relays to trip out. In contrast, the inlets open relatively quickly (i.e. within 2 minutes). An additional mode of operation is when the wind angle is significantly offset from a particular facade; flow predictions are more difficult to make in this situation.

The experiments set out to demonstrate the above when the appropriate climate conditions took place and the methodology is described in the next section.

### 4.13.2 Experimental procedure

Air speed sensors were placed in four positions in each stack and were mounted at heights of 9.5 m above floor level. They were also positioned at seven locations within the theatre, one on either side in the first row and two on the fifth row (LHS) with three in the corresponding row on the RHS. Sensors were placed at ankle height 0.5 m from inlet grilles.

External, wall and ceiling slab temperatures were monitored with thermistor sensors connected to Squirrel loggers. Temperatures were also recorded using the BMS The intake dampers were fully opened using the BMS, while the outlet configurations were varied such that at any particular time two of the four sets of windows at the top of each chimney were open while the other two sets were closed.

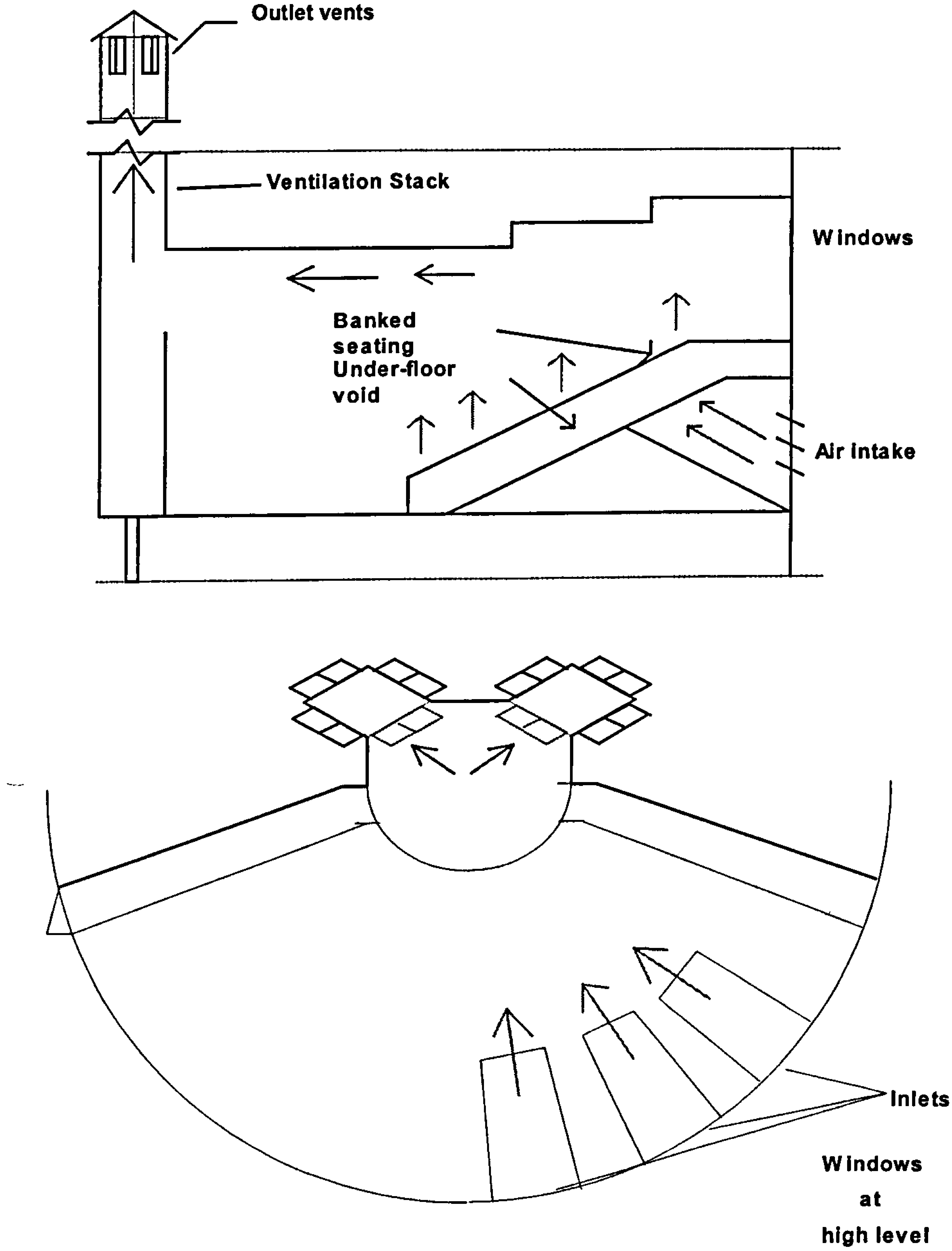



Figure 4.27 Section and plan of auditorium showing positions of inlets and outlet vents.

The sequence of opening positions used were as follows:

S1: SE/NE windows open, SW/NW windows closed;

S2: SE/NE windows closed, SW/NW windows open;

S3: SE/SW windows open, NE/NW windows closed;

S4: SE/SW windows closed, NE/NW windows open.

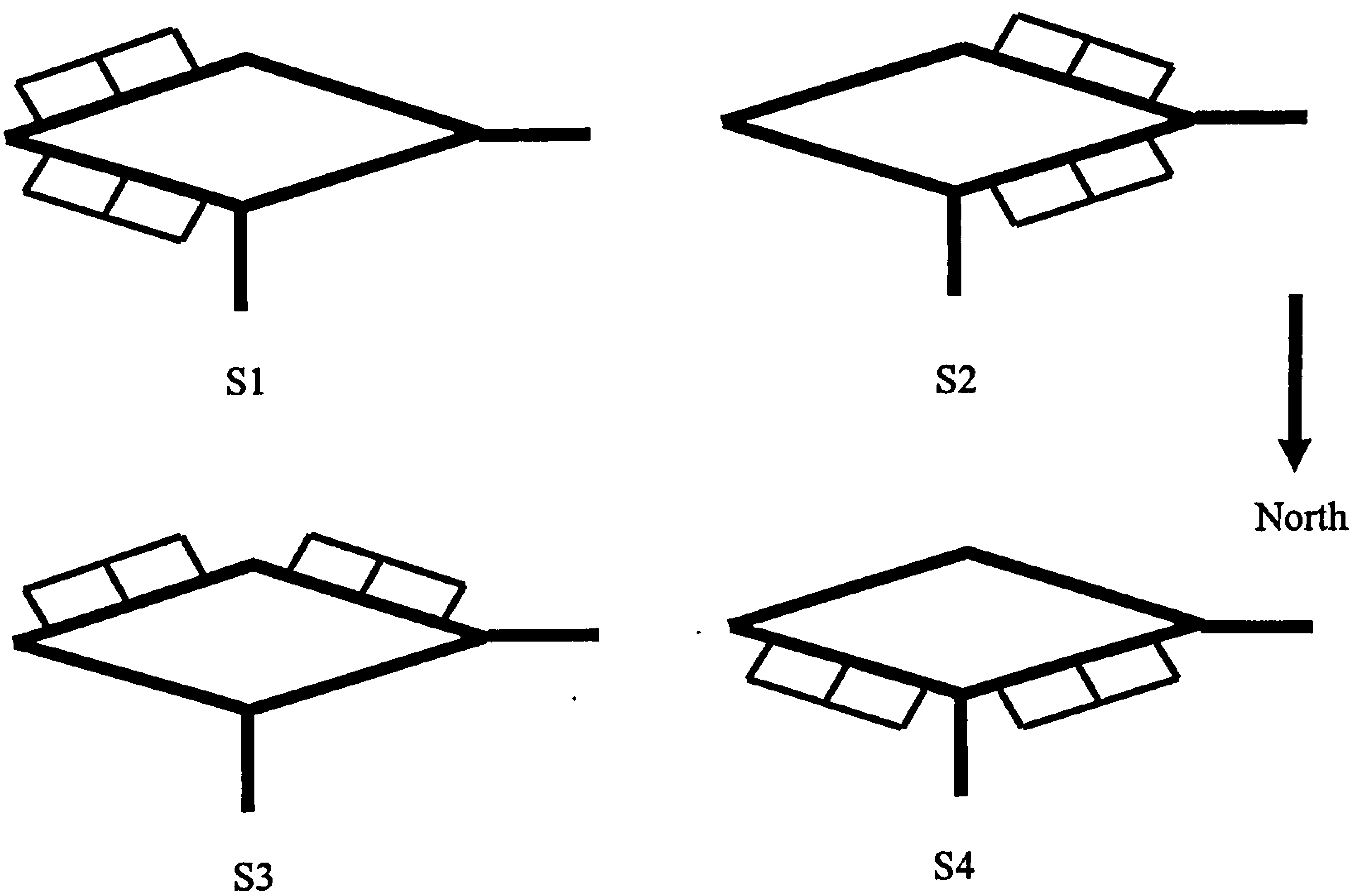



Figure 4.28 Positions of outlets at tops of stacks

The settings shown would provide openings on two facades of the stacks such that any one configuration would allow negative pressures (relative to the average internal pressure) to be generated on the facade facing into the wind with positive pressures being produced on the opposite facades (with openings fully extended).

Outlets were activated using manually operated knobs mounted within each stack at a height of approximately 1.5 m. Under normal operation via the BMS, they cannot open simultaneously.

This avoids overloading the supply circuits. Each actuator motor is operated in sequence, with each motor being switched on for about 2 minutes. The entire opening operation (outlets) takes around 8 minutes, with the LHS motors being energised before those on the RHS. This operation takes somewhat longer using manual control since the controls are only accessible by climbing into each stack and operating appropriate switches.

Weather conditions were monitored using two external temperature sensors and values of wind speeds plus directions were recorded at hourly intervals at an airfield weather station located approximately 9 miles (15 km) from Leicester city centre.

### 4.13.3 Results

Previously obtained results which examined the effect of temperature differences on flow rates indicated a good correlation during "winter" and "mid-season" periods (see figure 4.18). However, relatively large flow rates were evident even when temperature differences were small (i.e. less than 3 °C) or were negligible.

Measurements obtained during the "summer" showed that air speeds through the stacks were highly variable, due to the rapidly fluctuating wind speeds impinging onto the ventilation openings (see figure 4.28).

MISSING

PAGES

NOT

AVAILABLE

| Date (1995) | Tint (°C) | Text (°C) | Wind sp. (m/s) | Wind dir'n | Outlets Leeward ($m^3/s$) 1 | Outlets Windward ($m^3/s$) 2 | LHS ($m^3/s$) 3 | RHS ($m^3/s$) 4 |
|---|---|---|---|---|---|---|---|---|
| 22-Jun | 18.5 | 19.2 | 1.2 | NNW | 3.73 | 2.6 | 4.07 | 2.56 |
| 22-Jun | 20.9 | 22.3 | 2.2 | NNE | 2.2 | 2.51 | 1.88 | 2.01 |
| 03-Jul | 18.1 | 19.4 | 1 | E | 4.1 | 3.8 | 4.3 | 3 |
| 05-Jul | 19.3 | 22.0 | 2.2 | WSW | 4.1 | 2.4 | 2.8 | 3.2 |
| 05-Jul | 22.0 | 28.7 | 3.1 | WSW | 3.4 | 1.7 | 2.3 | 1.7 |
| 12-Jul | 22.5 | 24.3 | 1.4 | SW | 3.4 | 2 | 2 | 2.7 |
| 12-Jul | 24.3 | 27.2 | 1.4 | SW | 3.2 | 2.2 | 1.8 | 1.8 |
| 14-Aug | 23.1 | 23.8 | 2 | WNW | 4.1 | 2.8 | 3.2 | 5.1 |
| 15-Aug | 24.9 | 29.4 | 2 | WNW | 1.7 | 1.3 | 1.2 | 2.3 |
| 16-Aug | 25.0 | 29.8 | 1.2 | NNE | 1.8 | 1.4 | 1.7 | 1.4 |
| | | | | **Average** | **3.17** | **2.27** | **2.53** | **2.58** |

Table 4.2. Flowrates for different outlet configurations

The above data do not indicate a direct correlation between wind speeds and flow rates. However data obtained on a later date showed that higher wind speeds produced higher stack air velocities (i.e. greater bulk flow rates) (see figures 4.29 and 4.30). Buoyancy driven flow rates were almost constant throughout the period indicated, i.e. the internal and external temperatures changed by less than 1.0 °C from their average values.

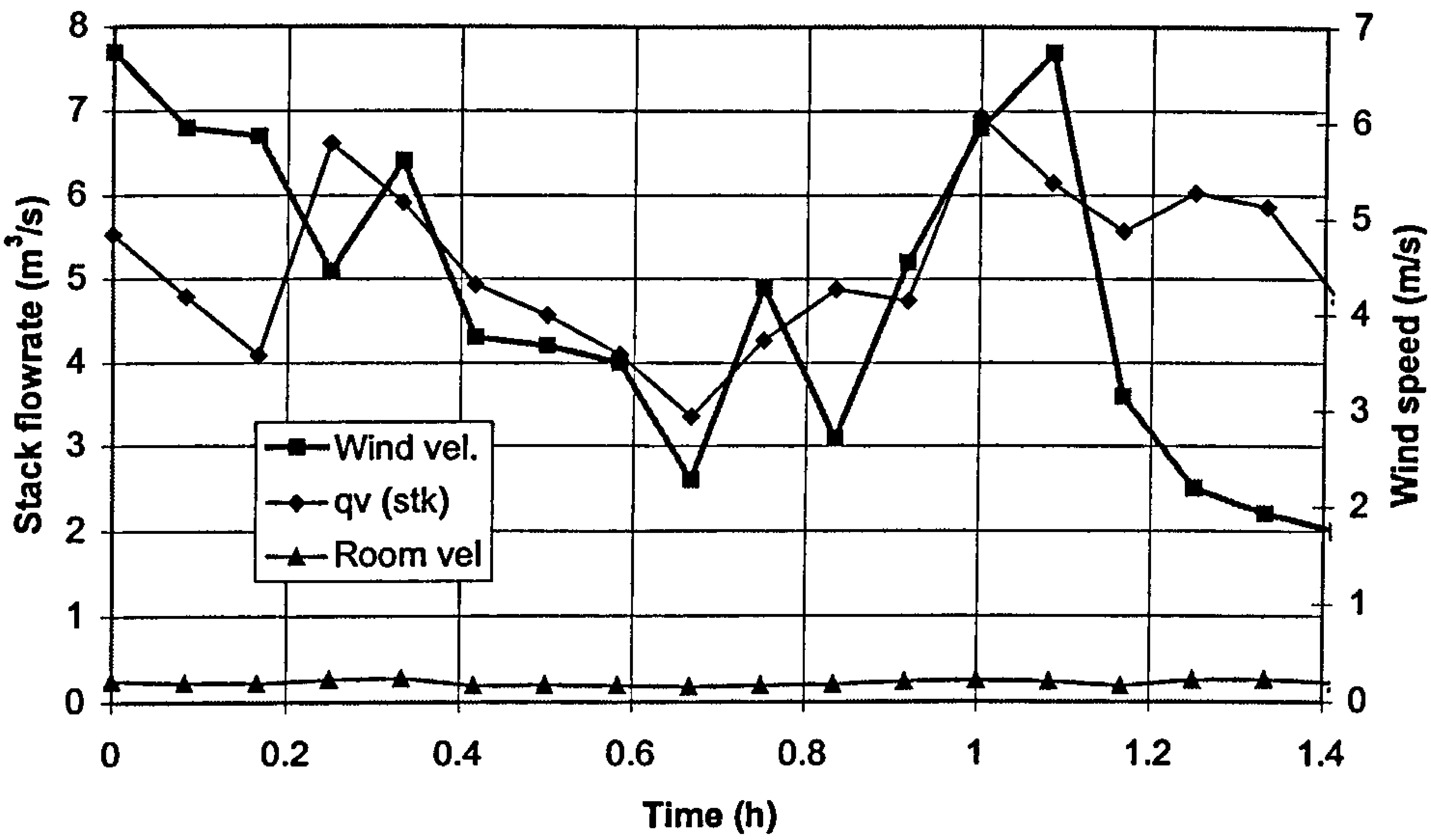



Figure 4.30 Wind speeds and ventilation rates (wind direction is WSW)

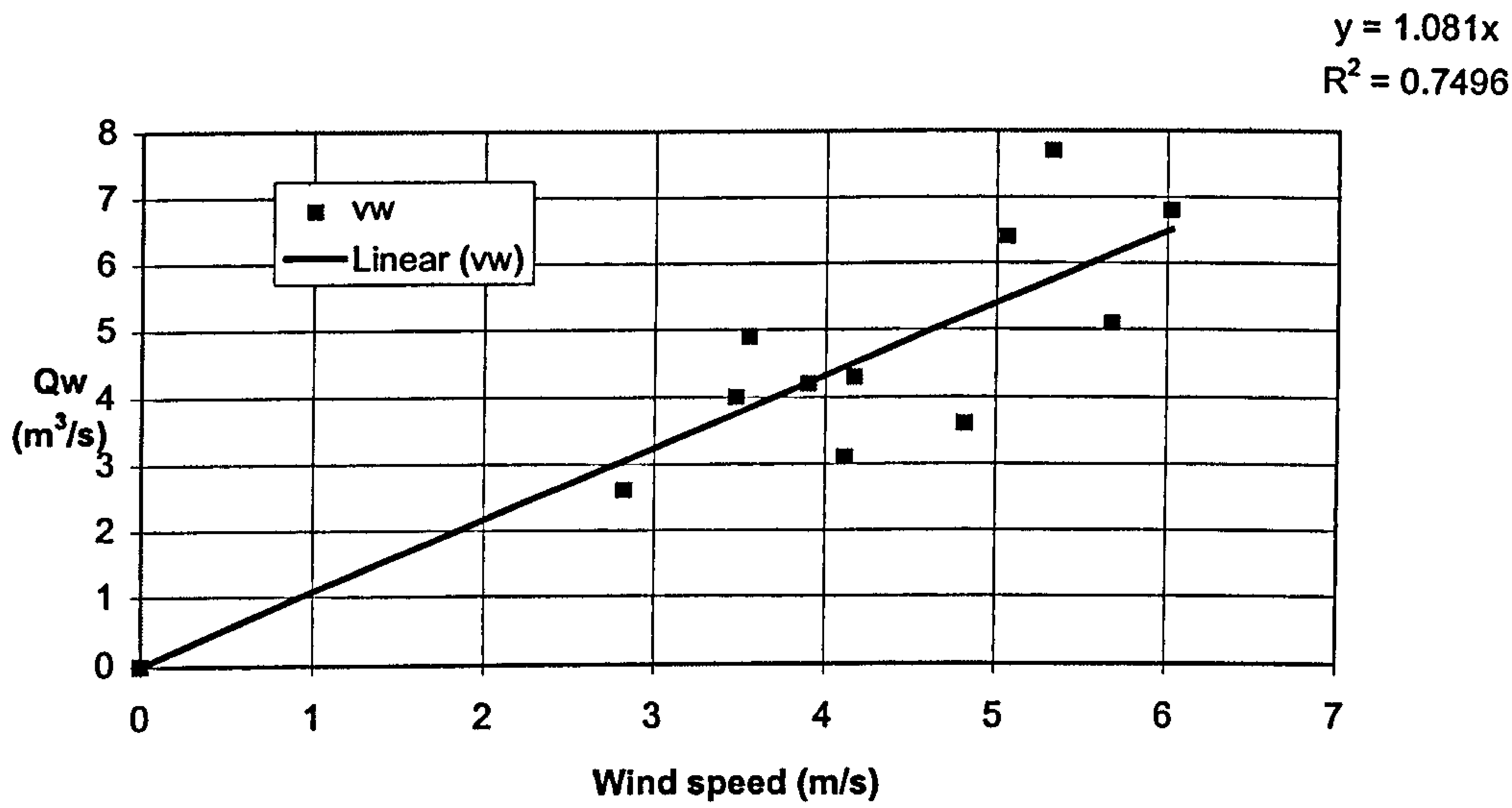



Figure 4.31 Wind speed versus flow rate (wind direction is WSW)

### 4.13.4 Conclusions

The ventilation stacks were sized for buoyancy driven flow only (such that enough fresh air would be provided even for a temperature difference of 3 °C between inside and outside). However substantial upflows were generated when external temperatures were equal to or greater than internal temperatures (i.e. wind induced pressures were frequently large enough to produce adequate flow rates and to overcome negative buoyancy effects).

Higher flow rates were measured when the leeward facing outlets were open with windward facing outlets closed. However the particular outlet vents used were not designed to enhance wind induced flow rates. If their shapes had been optimised using the results of wind tunnel tests or CFD analyses, then higher ventilation rates would have been achieved, or smaller vent areas and chimney dimensions could have been used to produce the required flow rates.

## 4.14 Analysis of wind speeds and directions

Key data which needs to be available during the design of a naturally ventilated building is reliable wind speed and direction data. If possible, this data should be available for several "typical" years. Other data required would include external temperatures, solar irradiances, and relative humidities. Data on local topography is also useful, this will determine the effects surrounding buildings have on wind speeds.

Data on wind speed and direction was originally available from two sources, East Midlands Airport and the Geography Department, Loughborough University. At a later date, a weather station was installed on top of Gateway house, which is a building located a few hundred metres away from the Queens Building. Data was also available via a weather station mounted on one of the Queens Building stacks but this proved to be highly unreliable when compared to the data obtained from East Midlands Airport (see figures 3.11 and 3.12).

It is useful to find the average and most common wind speeds and directions. The most common values can be determined from figures 4.31 and 4.32. These have been compiled from data files obtained from the Gateway House weather station from November 1995 to July 1996. The average wind speed over this period was 3.2 m/s at 26 m or 2.3 m/s at 10 m and the average wind direction was 171 degrees (SSE).

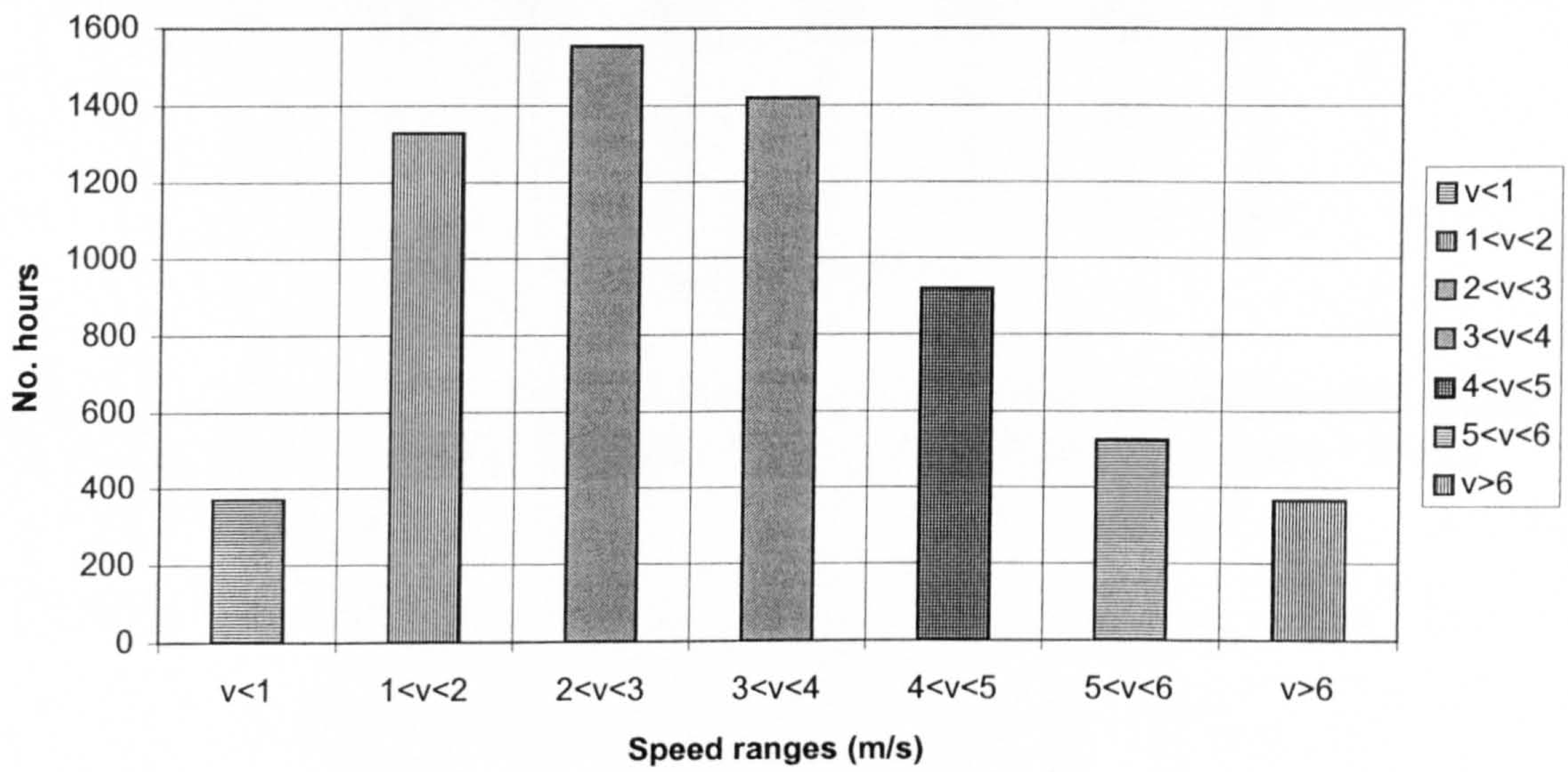



Figure 4.32 Histogram of the frequency of occurrence of particular wind speed ranges at De Montfort University from November 1995 to July 1996 (height of sensor = 26 m).

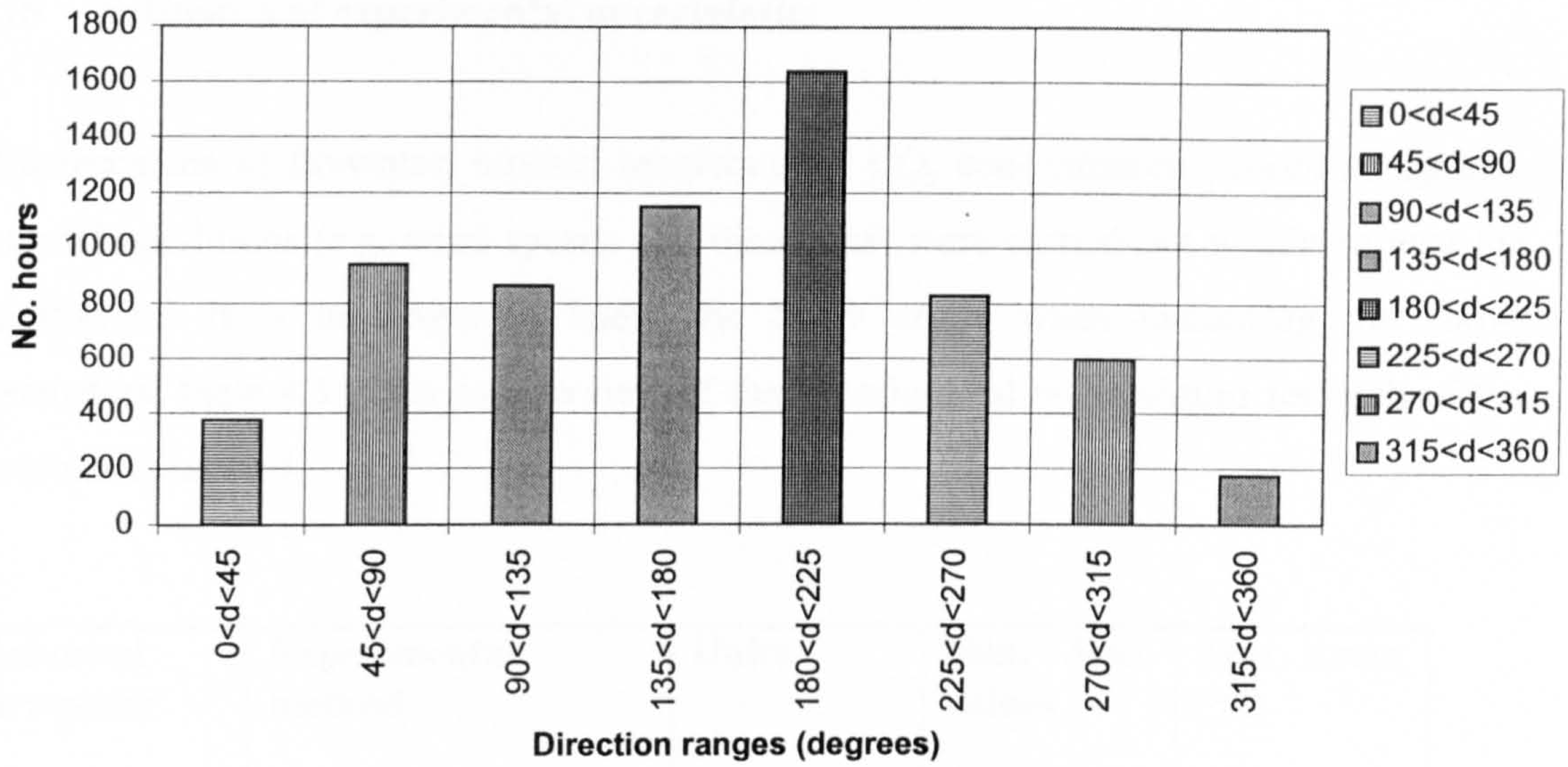



Figure 4.33 Histogram of the frequency of occurrence of particular wind direction ranges at De Montfort University from November 1995 to July 1996 (height of sensor = 26 m).

## 4.15 Estimates of experimental uncertainties

Measurements of flowrates, internal temperatures, $CO_2$ concentrations, room air speeds, external conditions (e.g. wind speeds and directions) were carried out within or near the Auditorium. It is important to know the likely errors when measuring the above parameters; table 4.3 gives an overview of the experimental uncertainties for each of the variables measured.

| **Measured parameter** | **Experimental method** | **Units** | **Min. - Max values** | **Max. Error (%)** |
|---|---|---|---|---|
| Flowrate | Tracer gas decay | $m^3 s^{-1}$<br>$h^{-1}$ | 0.03 to 10.7<br>0.1 to 36.5 | 10 |
| | Velocity probes | $m^3 s^{-1}$ | 4.48 to 8.65 | 10 |
| Infiltration | Tracer gas decay | $m^3 s^{-1}$ | 0.03 to 0.23 | 10 |
| Air speed | Omni directional | $m s^{-1}$ | 0.05 to 0.5 | 5 |
| (room) | Ultrasonic sensors | $m s^{-1}$ | 0.05 to 0.8 | 5 |
| | Heated thermistor | $m s^{-1}$ | 0.05 to 0.5 | 10 |
| (wind) | Cup anemometer | $m s^{-1}$ | 0.5 to 13.4 | 5 |
| Temperature | Thermistor probes (air temperature) | °C | 12.0 to 29.8 | 1 |
| | Thermistor probes (surface temperature) | °C | 17.0 to 26.0 | 1 |
| Pressure | Differential pressure sensor | Pa | 1.0 to 10.0 | 20 |
| Gas concs. | Infra red sensor (SF6) | ppm | 1.0 to 100.0 | 10 |
| | Infra red sensor ($CO_2$) | ppm | 300.0 to 1,500 | 10 |

Table 4.3 Experimental errors

Errors in the flowrate measurements using the velocity probes were minimised by measuring the chimney cross sectional areas accurately and by orientating the probe heads at right angles to the air flow. It is important that the probes are recalibrated annually. Errors using the tracer gas measurement were reduced by increasing the sample frequency and by ensuring that the initial gas concentration was reasonably uniform.

Temperature errors were reduced by using more expensive probes or by calibrating available sensors against detectors of known accuracy. It was important to be able to measure surface temperatures accurately as very small variations occurred in these values.

# 5 Comparison of previously completed experimental results with computer and physical models

## 5.1 ESP model

A dynamic thermal simulation was carried out by researchers at De Montfort University using ESP to see how internal temperatures would vary with time for particular occupancy heat gains, opening sizes, and external temperatures [4]. The following assumptions were made: 1) the opening were constantly open; 2) a stack height difference was 13.3 m; 3) the temperature differential between inside and outside was about 3 °C; 4) no wind induced flow was considered and 5) the room was occupied continuously for 8 hours a day, 5 days a week.

The above assumptions were made to make a difficult simulation somewhat simpler. However, in practise, only one of the assumptions is valid. The vents are designed to open only when there is a need for ventilation and when night venting is taking place. The stack height difference is 13.7 m (similar to the above figure). The temperature differential in summer can vary between 0 ° and 9 °C and negative differentials can occur. Wind induced forces appear to have a strong influence on flow rates in the summer. As is the usual situation in academic lecture theatres, the room is occupied intermittently and there are frequently several hours between lectures.

The model conditions can be taken as "worst case" scenarios so a conservative design should result.

An air-change rate of 6 was calculated for a indoor outdoor temperature difference of 3 °C (27 °C - 24 °C). The effective area was 1.77 $m^2$. If the equivalent flow rate is increased by the ratio of the actual to the model vent areas (2.02) and a function of the different temperature difference (23.5 °C - 21.6 °C), one resulting value is 9.7 ach (1/hr). This can be compared to the actual measured value of 12.0 (1/hr). Another run (with temperatures of 22.8 °C and 21.7 °C for inside and outside respectively) gave a flow rate of 15.7 ach (1/hr) which compares poorly with a calculated value (corrected as before for area and temperature) of 7.4 ach (1/hr). The differences could be accounted for by flow rates induced by wind (which was not calculated by the ESP model).

| $T_{ai}$ (°C) | $T_{ao}$ (°C) | ($T_{ai}$ - $T_{ao}$) (°C) | Flow rate ($m^3/s$) (ESP) | (actual) |
|---|---|---|---|---|
| 23.5 | 21.6 | 1.9 | 2.84 | 3.52 |
| 22.8 | 21.7 | 1.1 | 2.17 | 4.60 |

Table 5.1 comparison of flow rates found using ESP and measured flow rates

## 5.2 Salt bath model

Steady state conditions in the space were also modelled by research staff at Cambridge Environmental Research Consultants Ltd. using a salt bath model, where a 1/75 scale perspex model was built and inverted in a tank of water [5]. Heat sources (equivalent to 15 kW) were modelled using salt solutions of varying concentrations. These solutions were dyed to aid visualisation. The movement of the solutions was photographed with a video camera for particular vent areas and "heat" gains.

The assumptions were as follows: 1) As stated, the model examined steady state conditions only; 2) a temperature difference between inside and outside was 5.8 °C; 3) wind effects were not modelled; and 4) heat transfer to/from internal surfaces was not considered.

Displacement flow was observed, with a clearly defined layer interface between low and high level occurring at approximately the level of the lower edge of the chimney openings (i.e. 4.5 m). However inflow took place through "windows" (when inlets, stack outlets and windows were open) with the incoming "air" falling into the space and mixing with the room air (thus mixed flow was observed for this situation).

A flow rate of 2.2 $m^3/s$ (9.1 ach) was calculated for steady-state flows, for a temperature difference of 5.8 °C and inlet and exhaust (stack) opening areas of 5.2 and 2.6 $m^2$. As for the ESP model, this is somewhat less than values measured when a similar temperature difference occurred (see above).

# 6 Developing a model of environmental conditions in the auditorium

## 6.1 Introduction

As stated previously, flow rates are driven by both buoyancy forces and wind induced pressures. For much of the year, it is sufficient to assume that buoyancy forces only are operating. However, when temperature differences are 3 °C or less, or when external temperatures are greater than internal temperatures, then it is necessary to consider the magnitude of wind induced pressure differences.

A ventilation model will be derived. This will predict bulk flow rates, assuming ventilation rates can be found independently of air temperatures (i.e. if internal air temperatures are allowed to be constant over the period of interest). The effects of reducing internal temperatures on flow rates can then be considered.

Two thermal models will be derived. The first model will predict internal air temperatures, once the building fabric cooling and heating time constants have been found. The time constants will be the times taken for the variables to rise (or drop) to 0.632 of their steady-state value and will be found by linear interpolation. The second model will predict internal temperatures in summer. The derivation of the models assumes that there is one major factor affecting internal temperatures (i.e. the output of heating system in winter and the fluctuations in external temperature in summer). A criteria will need to be established which will determine which model is to be used for given external conditions.

Once the particular models have been produced and checked, then they can be compared to experimental values. If there is a reasonable degree of correspondence between models and measurement data (i.e. errors of +/- 15 %) then predictions can be made of the internal environment of the auditorium in terms of air flow, air speeds, and temperatures. It may then be valid to predict conditions in similar type of spaces, i.e. rooms with approximately similar time constants, stack heights, opening areas and climate types.

Models are required to predict the following for "any" set of external conditions:

- Ventilation rates
- Internal and surface temperatures

These models will need to be used together, i.e. the above parameters are interrelated. Internal and external temperatures can be used to predict ventilation rates.

Once the above models have been produced and validated, they can be used to determine the internal environmental conditions in the auditorium (and in enclosures of similar size, geometry and thermal mass) and at any time of the year. If the average internal temperature can be predicted, then it should be possible to estimate heating energy use (i.e. the heating system is "on" if the average temperature is more than 0.5 °C below the set point and is "off" if the average temperature is at or above the set-point).

The use of the models will allow "optimum" control strategies to be selected; i.e. what strategy minimises the variation of internal temperature about a set point, and provides adequate ventilation while still maintaining comfortable conditions in the space and minimising heating energy use.

## 6.2 Prediction of ventilation rates and air speeds

### 6.2.1 Buoyancy driven flow only (winter and mid-season conditions)

As was stated earlier, the CIBSE stack equation was used at the design stage to determine whether a set of given opening areas would be big enough to provide enough fresh air. The results have shown that sufficient air could be provided to the space when the vents were open and that good agreement was reached between the flow rate calculated by the CIBSE formula and the measured flow rate. However this assumes that the stack effect is the predominant driving force. It appears that for a temperature difference of less than c. 2 °C wind induced pressure differences are approximately the same as those generated by the stack effect, assuming the wind speed is about 3 m $s^{-1}$. So the formula gives satisfactory flow rate predictions in "winter" and mid-season" but less accurate predictions in "summer" (see figures 6.1(a) and 6.1(b)).

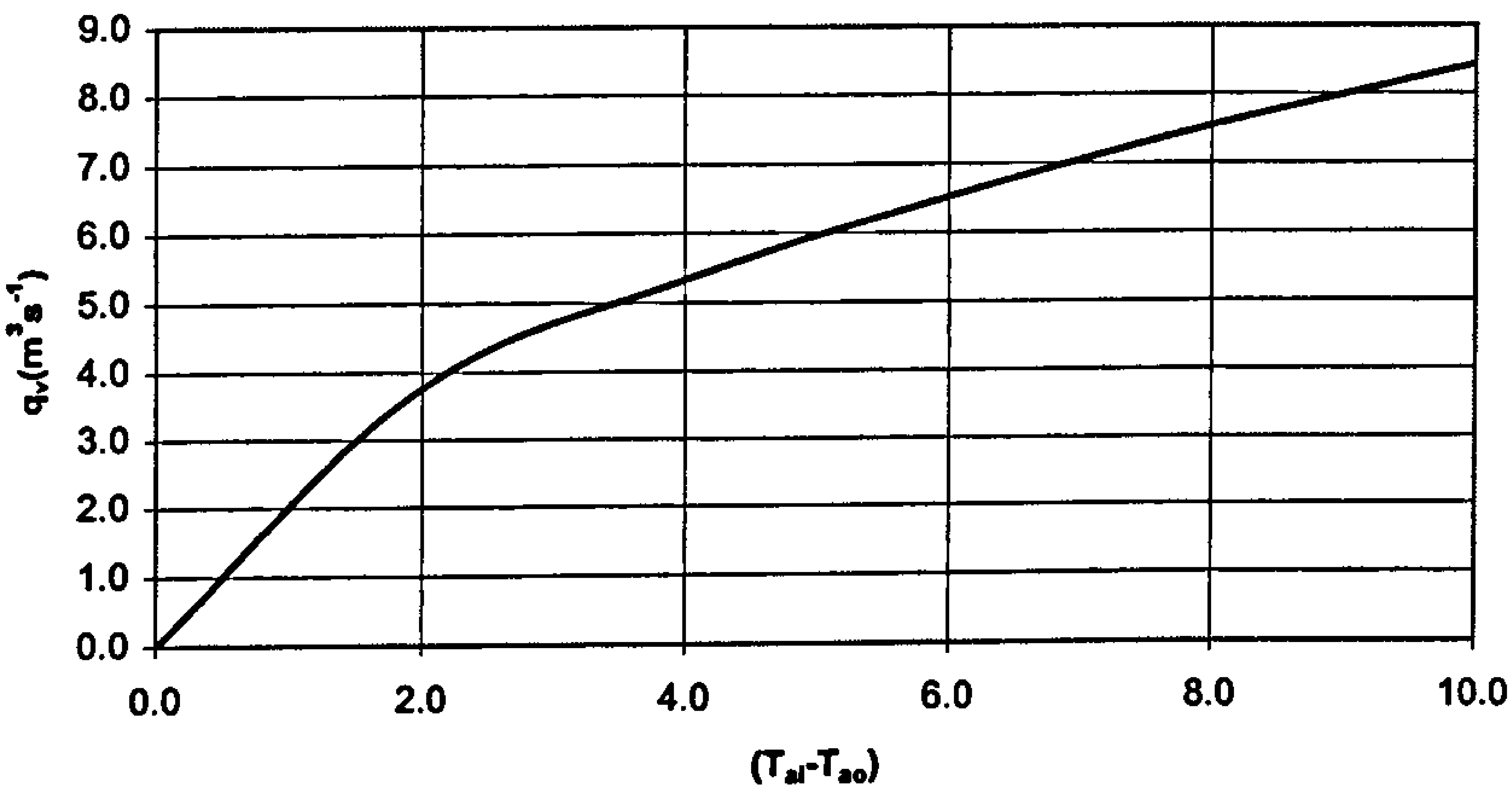



Figure 6.1(a) Flowrate versus inside minus outside temperature using a simple ventilation model

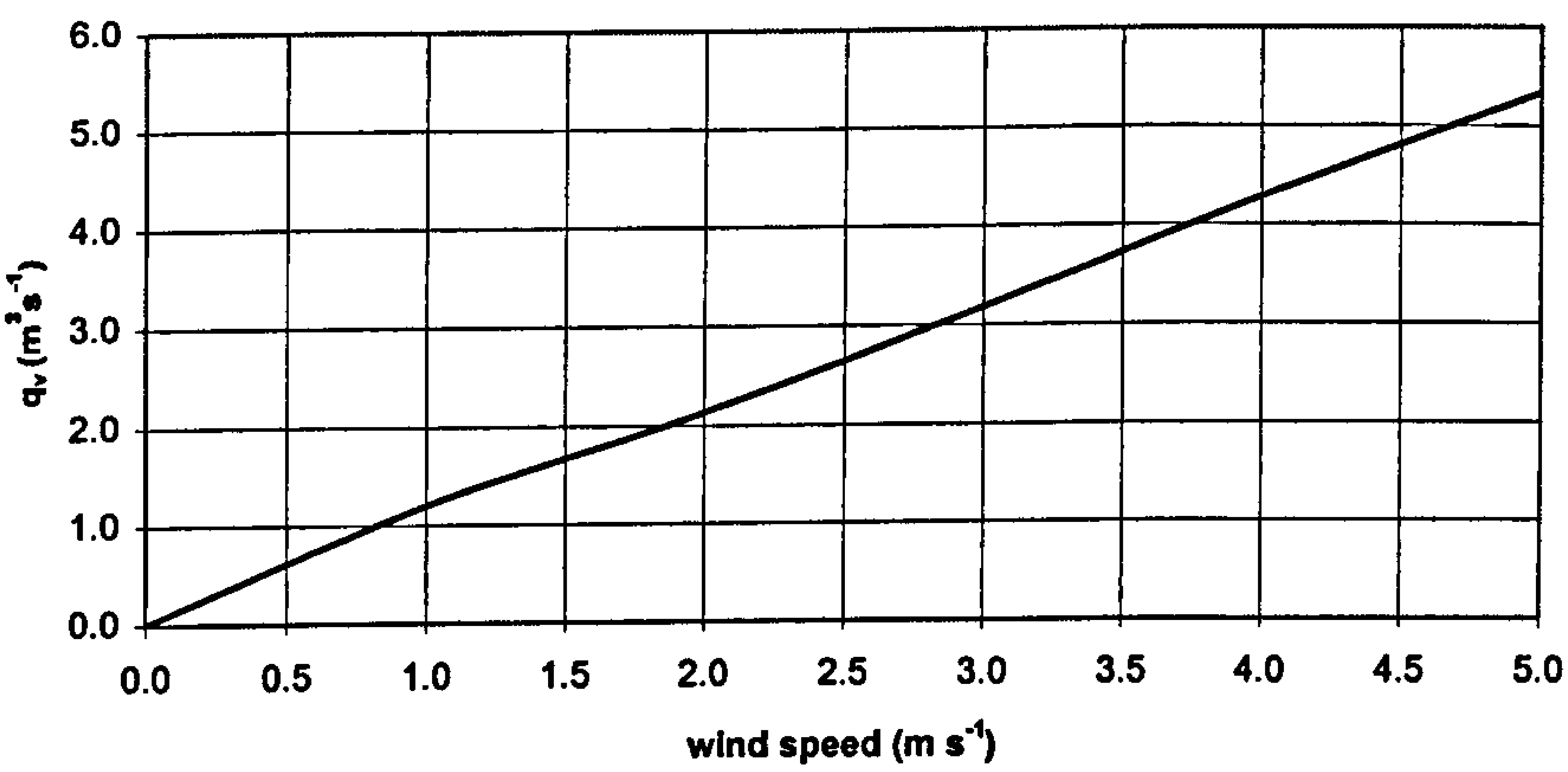



Figure 6.1(b) Flowrate versus wind speed using a simple ventilation model

Data shown in figures 6.1(a) and 6.1(b) have been obtained using a simple single zone ventilation model which calculates buoyancy and wind driven flowrates using an iterative procedure.

A formula for deriving stack induced pressures has been produced (see equation number 9 in Chapter 1). This can be incorporated into Bernouillis equation to find flowrates, i.e.;

$$q_v(m^3/s) = C_d A_E \sqrt{\frac{2\Delta P}{\rho_h}} \qquad (6.1)$$

where $C_d$ = discharge coefficient;

$A_E$ = Effective opening area ($m^2$);

$$\Delta P = \frac{P_a}{RT_i}\left[\frac{T_{ai} - T_{ao}}{T_{ao}}\right] g(h_2 - h_1) \qquad (6.2)$$

where $P_a$ = Atmospheric pressure (Pa);

R = gas constant for air (=287 J $kg^{-1}K^{-1}$);

$T_{ai}$ = average internal temperature (°C);

$T_{ao}$ = average external temperature (°C);

g = 9.81 m $s^{-2}$;

$h_2$ = height of outlet centroid;

$h_1$ = height of inlet centroid;

$\rho_h$ = internal air density.

Equation 6.1 can be re-written as:

$$q_v(m^3/s) = C_d A_E \sqrt{2g(h_2 - h_1)\left[\frac{T_{ai} - T_{ao}}{T_{ao}}\right]} \qquad (6.3)$$

Assuming that very little variation occurs in $h_2$ and $h_1$ for different opening areas, then for a given opening setting, the above equation can be simplified to:

$$q_v(m^3/s) = k_{os}\left[\frac{T_{ai} - T_{ao}}{T_{ao}}\right]^{0.5} \qquad (6.4)$$

where $k_{os}$ = stack ventilation constant for a particular os (outlet setting) (i.e. for settings of 50 %; 75 % and 100 %).

The outlet setting values can be found by plotting $((T_i\text{-}T_o)/T_o)^{0.5}$ versus measured volumetric flowrates (see figures 6.2, 6.3 and 6.4). For each graph, a "best fit" line has been drawn through the points and the slope of this line has been calculated.

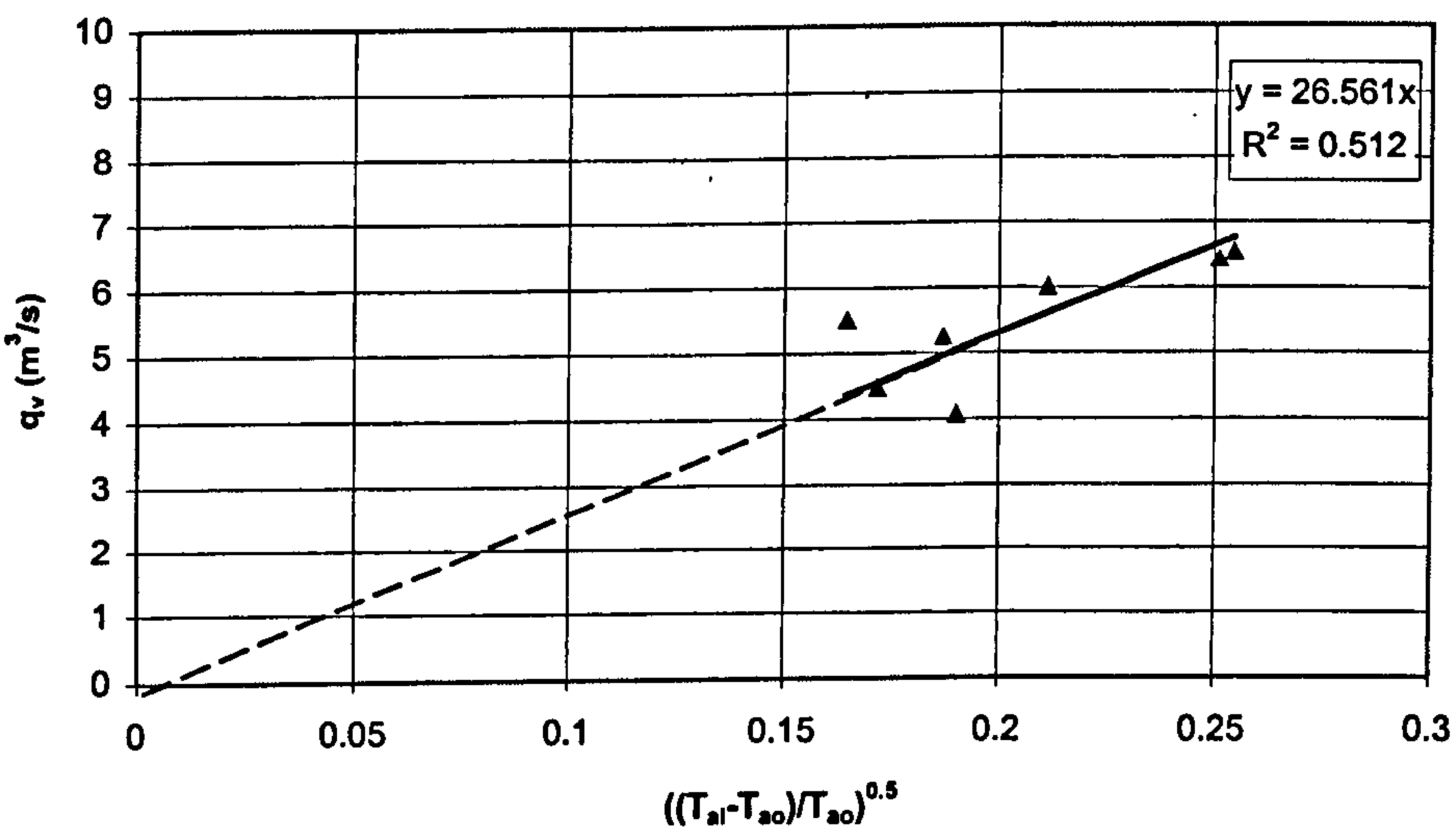



Figure 6.2 Measured flowrate versus $((T_{ai}\text{-}T_{ao})/T_{ao})^{0.5}$. Effective area is 68 % of maximum area.

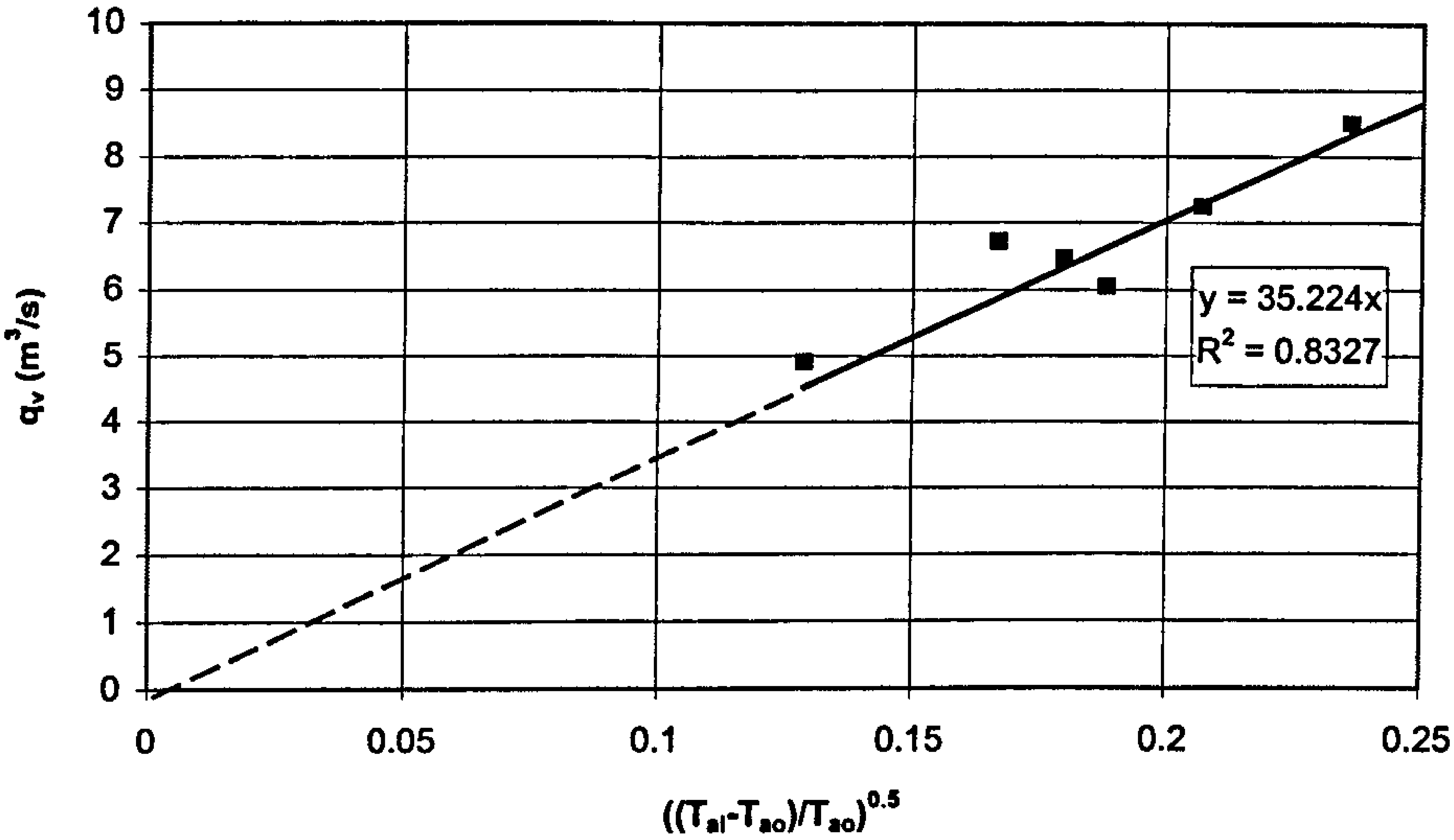



Figure 6.3 Measured flowrate versus $((T_{ai}-T_{ao})/T_{ao})^{0.5}$. Effective area is 89 % of maximum area.

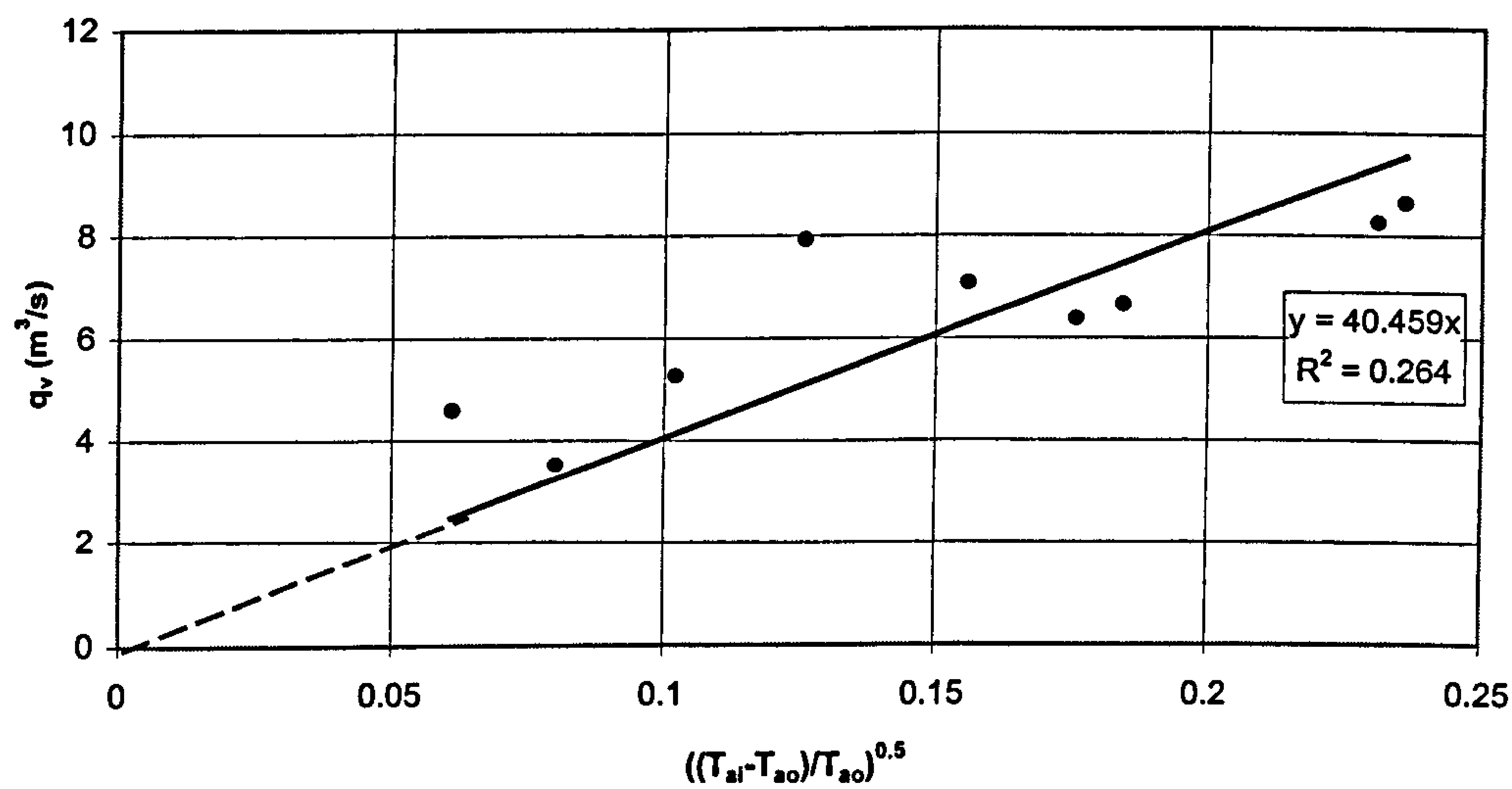



Figure 6.4 Measured flowrate versus $((T_{ai}-T_{ao})/T_{ao})^{0.5}$. Effective area is 100 % of maximum area.

The above plots have allowed the slopes of best fit lines (or $k_{0s}$ coefficients) to be obtained. Hence the coefficients are as follows:

$k_{50} = 26.56\ m^3/s$
$k_{75} = 35.22\ m^3/s$
$k_{100} = 40.46\ m^3/s$

It is interesting to compare the above coefficients with those found using equation 6.4. The coefficient term is found by comparing equation 6.3 and equation 6.4, i.e.:

$$k_{os(calc)}(m^3/s) = C_d A_E \sqrt{2g(h_2 - h_1)} \qquad (6.5)$$

Substituting realistic or actual values for $C_d$, g, $h_2$, and $h_1$ gives (i.e. let $C_d = 0.6$, $h_2 = 18.5$ m, $h_1 = 4.8$ m) :

$$k_{os(calc)} = 9.84\ A_E \qquad (6.6)$$

Using previously calculated values for effective areas, this gives:

$k_{50} = 9.84\ (2.43) = 23.90\ m^3/s$ (BMS setting of 50 %)
$k_{75} = 9.84\ (3.18) = 31.28\ m^3/s$ (BMS setting of 75 %)
$k_{100} = 9.84\ (3.57) = 35.12\ m^3/s$ (BMS setting of 100 %)

If the constants $k_{50}$, $k_{75}$ and $k_{100}$ are divided by the corresponding experimentally found values (see figures 6.2, 6.3, and 6.4 and table 6.1), the values 0.90, 0.89 and 0.87 are obtained. This implies that (i) the stack formula used is applicable and (ii) that the effective areas are under-predicted by the derived formula.

| Opening setting (%) | Effective area ($m^2$) | Coefficient ($k_{os}$) ($m^3$/s; measured) | Coefficient ($k_{os}$) ($m^3$/s; calculated) | $k_{os(calc)}$ /$k_{os(meas)}$ |
|---|---|---|---|---|
| 50 | 2.43 | 26.56 | 23.90 | 0.90 |
| 75 | 3.18 | 35.22 | 31.28 | 0.89 |
| 100 | 3.57 | 40.46 | 35.12 | 0.87 |

Table 6.1 Comparison of experimentally derived and calculated stack coefficients

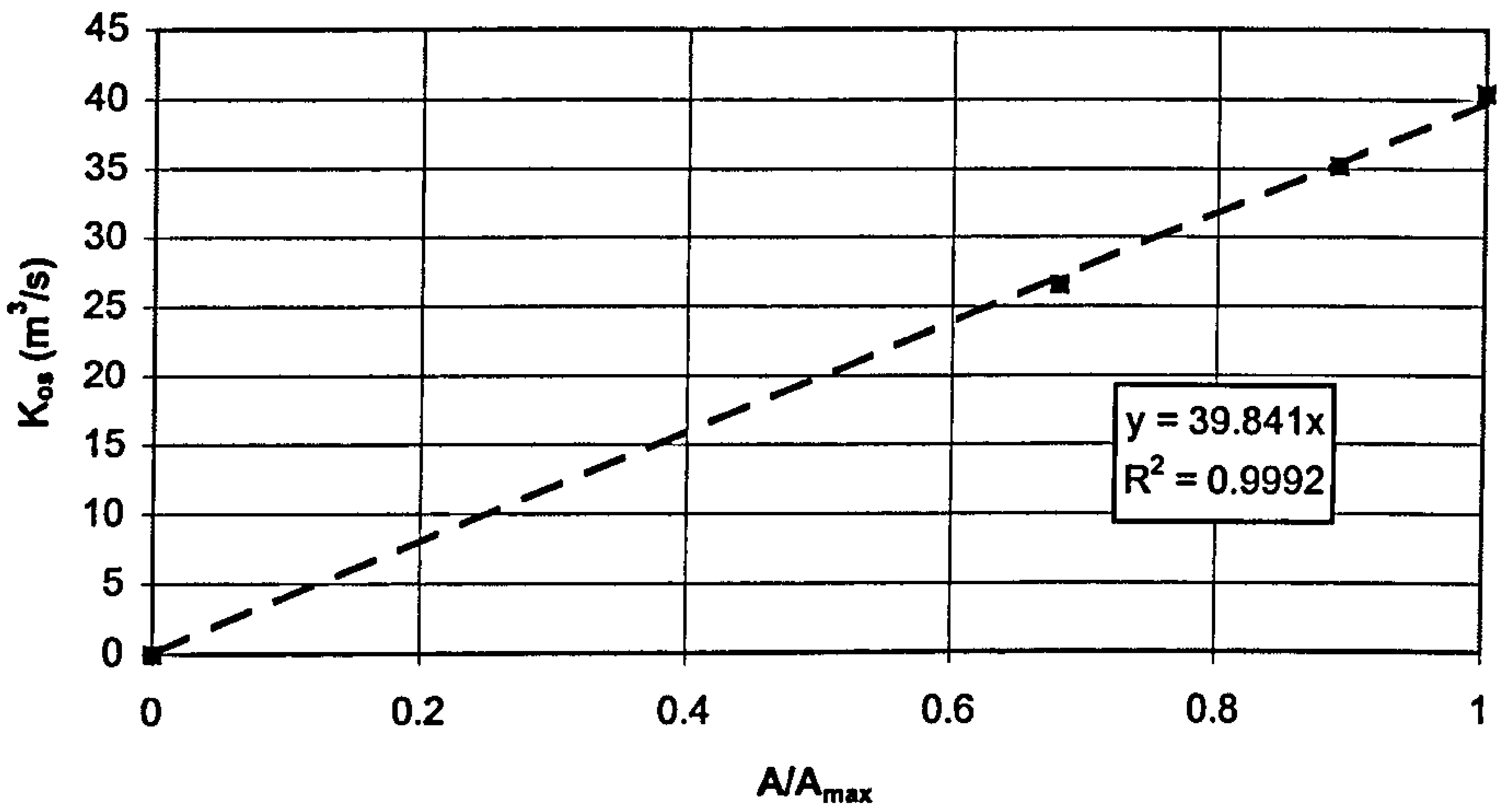



Figure 6.5 Stack flow coefficients versus area ratios

If the flow coefficients are plotted against area ratios (i.e. the ratio of the actual effective area used to the maximum area obtainable) then one coefficient can be found which is inversely proportional to the effective the area used. Thus equation 6.3. can be re-written as:

$$q_v(m^3/s) = 39.8\left[\frac{T_{ai} - T_{ao}}{T_{ao}}\right]^{0.5} \qquad (6.7)$$

The flow co-efficient is then found from:

$$k_{os} = \frac{39.8}{A_E} \tag{6.8}$$

(all terms shown have been defined previously).

The following tables compare measured and calculated flow rates for selected winter and mid-season runs and gives the percentage difference between the two sets of figures. Calculated flowrates have been found using the experimentally derived flow coefficients ($k_{50}$; $k_{75}$ and $k_{100}$; see above) and equation 6.3. Good agreement is evident for most runs.

| Season | BMS Settings (%) | Opening Area ($m^2$) | Avg. Temp. Internal (°C) | External (°C) | Measured Flowrate ($m^3/s$) | Calculated Flowrate ($m^3/s$) | Difference (%) |
|---|---|---|---|---|---|---|---|
| | | | | | | | |
| Winter | 50 | 2.43 | 21.2 | 3.3 | 6.53 | 6.76 | -3.5 |
| | 75 | 3.18 | 20.5 | 3.2 | 8.26 | 8.82 | -6.8 |
| | 100 | 3.57 | 19.8 | 4.4 | 8.64 | 9.55 | -10.5 |
| | 50 | 2.43 | 19.5 | 2.2 | 6.43 | 6.67 | -3.7 |
| | 75 | 3.18 | 19.7 | 4.3 | 8.50 | 8.32 | 2.1 |
| | 100 | 3.57 | 19.5 | 4.7 | 8.26 | 9.35 | -13.3 |
| | 50 | 2.43 | 17.3 | 4.9 | 6.00 | 5.62 | 6.4 |
| | 75 | 3.18 | 16.9 | 5.0 | 7.25 | 7.29 | -0.6 |
| Average | | | | | | | 7.1 |

Table 6.2.(a) Comparison of measured and calculated flow rates for selected winter season runs.

| Season | BMS Settings (%) | Opening Area ($m^2$) | Avg. Temp. Internal (°C) | External (°C) | Measured Flowrate ($m^3/s$) | Calculated Flowrate ($m^3/s$) | Difference (%) |
|---|---|---|---|---|---|---|---|
| Mid- | 50 | 2.43 | 19.9 | 10.0 | 5.24 | 4.97 | 5.1 |
| season | 75 | 3.18 | 19.2 | 9.2 | 6.05 | 6.63 | -9.5 |
| | 100 | 3.57 | 18.8 | 9.2 | 6.66 | 7.46 | -12.1 |
| | 50 | 2.43 | 20.4 | 12.0 | 4.48 | 4.56 | -1.8 |
| | 75 | 3.18 | 20.8 | 11.6 | 6.47 | 6.33 | 2.1 |
| | 100 | 3.57 | 20.4 | 11.6 | 6.38 | 7.11 | -11.5 |
| | 50 | 2.43 | 19.0 | 11.3 | 5.51 | 4.37 | 20.7 |
| | 75 | 3.18 | 18.8 | 10.9 | 6.73 | 5.88 | 12.7 |
| | 100 | 3.57 | 18.0 | 11.1 | 7.08 | 6.31 | 10.9 |
| | 50 | 2.43 | 20.2 | 14.5 | 7.95 | 3.74 | 52.9 |
| | 75 | 3.18 | 20.1 | 15.3 | 4.91 | 4.53 | 7.8 |
| | 100 | 3.57 | 20.0 | 15.4 | 7.92 | 5.09 | 35.8 |
| Average | | | | | | | 20.9 |

Table 6.2(b) Comparison of measured and calculated flow rates for selected mid-season runs.

Based on the data shown in table 6.2(a) and 6.2(b) it would be reasonable to conclude that accurate predictions (to 10 %) can be generated using the CIBSE stack formula provided the average inside/outside temperature difference is of the order of 10 °C. An average error of 9.1 % was obtained for winter and mid-season conditions with inside minus outside temperature differences from c. 8 °C to 15 °C.

### 6.2.2 Wind induced flowrates

Flowrates have been measured over several hours on a particular day (c. 10:00 to 17:00) together with inside and outside air temperatures, wind speeds and directions. From the data supplied, the stack flow rate can be readily calculated. The total flowrate is assumed to be the square root of the square of the stack flow rate plus the square of the wind driven flow rate. To test a simple relationship between wind speed and flow rate (for a given wind direction), the stack flow rate (estimated) is subtracted from the total (measured) flow rate. This gives the wind induced flow rate only and this can now be plotted against wind speed. However the correlation is not very satisfactory.

Results for one particular run showed a good correlation between wind speed (as measured at East Midlands Airport) and flowrate. However modelling is required to indicate the relationship for a range of wind directions.

### 6.2.2(a) Calculation of wind and stack induced flowrates

A simple single zone model has been used to predict flowrates as generated by the combined effects of wind and temperature difference. This involved assuming constant internal and external temperatures, and unchanging wind speeds and directions. The assumption of constant conditions is reasonable for short periods of time, i.e. 15 minutes.

Initially stack pressures are calculated using equation 6.1. The location of the neutral pressure line depends on the relative sizes of the openings and needs to be determined. This will allow values to be assigned to pressures at the opening height values.

Determination of wind induced pressures is more involved. Initially pressure coefficients have to be found by interpolating from available data ([20] and other references). Unfortunately this data is available only for a limited number of obstruction profiles (e.g. low rise square sectioned buildings) and is not very suitable for openings on chimney extracts with sharp edges (where flow fields are likely to be complex and highly dependent on wind direction).

However, despite the above limitations, pressure coefficient values were found for 4 separate wind directions [20]. Values of wind speed have been calculated based on an outlet height of 18.5 m; wind speed values for inlets are found by reducing the outlet values by the cube root of the ratio of the height differences. Thus a wind speed at 4.8 m above ground level is found from:

$$v_{4.8m} = v_{18.5m}\left(\frac{4.8}{18.5}\right)^{0.33} \qquad (6.9)\ [20]$$

$$v_{4.8m} = 0.64\ v_{18.5m}$$

The angles between the outward normals of the openings and a particular wind direction were calculated and used to find pressure coefficients. The following table gives a set of values obtained:

| | Opening number | | | | | | |
|---|---|---|---|---|---|---|---|
| Wind Dir'n | I1 | I2 | I3 | O1 | O2 | O3 | O4 |
| Angle (°) | 92 | 71 | 59 | 270 | 316 | 126 | 90 |
| WSW | -0.20 | -0.17 | -0.14 | -0.20 | -0.12 | -0.34 | -0.20 |
| Angle (°) | 259 | 238 | 216 | 67 | 113 | 293 | 247 |
| E | -0.41 | -0.53 | -0.58 | -0.14 | -0.48 | -0.14 | -0.48 |
| Angle (°) | 79 | 58 | 36 | 113 | 67 | 113 | 67 |
| W | -0.25 | -0.06 | 0.10 | -0.48 | -0.14 | -0.48 | -0.14 |
| Angle (°) | 79 | 100 | 122 | 90 | 136 | 45 | 91 |
| ENE | -0.25 | -0.41 | -0.53 | -0.35 | -0.6 | 0.06 | -0.35 |

Table 6.3 Angles between opening outward normals and wind directions with corresponding average pressure coefficients. (I1 to I3 are low level vents (inlets); O1 to O3 are high level vents (outlets)).

The wind pressure at each opening can now be determined from:

$$P_{w_i} = 0.5\rho_c Cp_i v_i^2 \qquad (6.10)$$

where $\rho_c$ = density of outside air (kg/m$^3$);

$Cp_i$ = average pressure coefficient at opening i;

$v_i$ = wind speed (m/s) at height of opening i

Stack pressures are calculated using:

$$\Delta P_{st} = 3463(h_n - h_i)\left(\frac{1}{T_{ao}} - \frac{1}{T_{ai}}\right) \qquad (6.11)$$

where $3{,}463 = \frac{P_a g}{R} = \frac{101{,}325(9.81)}{287}$

$h_n$ = neutral pressure line (m);

$h_i$ = Height of opening i (m);

$T_{ao}$ = Outside air temperature (K);

$T_{ai}$ = Inside air temperature (K).

From the above, total pressure differences at each opening can be estimated. To find mass flowrates, flow coefficients for each vent are established. These are obtained using an estimated discharge coefficient (set equal to 0.6) and the vent areas. It is assumed that all openings behave like "large" openings" (i.e. the flow exponent is set to 0.5).

Flow coefficients are found from:

$$k_i = C_d A_i \left(\frac{2}{\rho}\right)^{n_i} \qquad (6.12)$$

where $C_d$ = discharge coefficient (=0.6);

$A_i$ = opening area ($m^2$);

$\rho_j$ = Density of air coming from the lower pressure zone (kg $m^{-3}$);

$n_i$ = Flow exponent (=0.5).

A one zone model is being used, thus only one internal pressure must be determined. This internal pressure (for a given set of conditions) is found by iteration; i.e. initial values of internal pressure are assumed; and the total mass flow into and out of the zone is then found. If the total net flow (i.e. the sum of inflows and outflows) is not zero, then a new value of internal pressure is calculated. This new value should allow a lower total flow value to be determined. The internal pressure is altered until flow balance is achieved.

Results obtained are indicated in the following tables for 4 different wind directions selected and are compared to measured values. Error terms have also been found (these indicate the relative difference between measured and calculated volume flowrates):

| **Avg. Temp.** | | **Wind** | **Wind** | **Flowrate** | **Flowrate** | **Error (%)** |
|---|---|---|---|---|---|---|
| Internal (°C) | External (°C) | Speed (m/s) | Dir'n | ($m^3/s$) (meas.) | ($m^3/s$) (calc; s+w) | |
| 19.9 | 10.0 | 4.6 | WSW | 5.24 | 5.55 | -6.0 |
| 19.2 | 9.2 | 5.2 | WSW | 6.05 | 7.61 | -25.7 |
| 18.8 | 9.2 | 4.9 | WSW | 6.66 | 8.96 | -34.6 |
| 20.2 | 14.5 | 4.2 | WSW | 7.95 | 4.77 | 40.0 |
| 20.1 | 15.3 | 2.9 | WSW | 4.91 | 4.50 | 8.4 |
| 20.0 | 15.4 | 3.1 | WSW | 7.92 | 5.80 | 26.8 |
| 20.2 | 16.8 | 4.4 | WSW | 5.35 | 4.58 | 14.4 |
| 20.3 | 17.4 | 4 | WSW | 5.38 | 5.26 | 2.2 |
| 20.4 | 17.4 | 4.4 | WSW | 5.26 | 7.01 | -33.2 |
| | | | | | **Avg Error(%)** | **24.9** |

Table 6.4(a) Comparison of measured and calculated ventilation rates (wind direction is WSW)

| Avg. Temp. Internal (°C) | Avg. Temp. External (°C) | Wind Speed (m/s) | Wind Dir'n | Flowrate (m3/s) (meas.) | Flowrate (m3/s) (calc; s+w) | Error (%) |
|---|---|---|---|---|---|---|
| 24.6 | 24.9 | **2.8** | ENE | 1.6 | 3.57 | -121.5 |
| 25.0 | 25.9 | **2.8** | ENE | 2.3 | 3.26 | -40.8 |
| 27.5 | 29.8 | **3.4** | ENE | 1.2 | 3.17 | -163.8 |
| 22.8 | 21.7 | 2.6 | ENE | 4.6 | 4.16 | 9.6 |
| | | | | | **Avg Error(%)** | **104.1** |

Table 6.4(b) Comparison of measured and calculated ventilation rates (wind direction is ENE)

| Avg. Temp. Internal (°C) | Avg. Temp. External (°C) | Wind Speed (m/s) | Wind Dir'n | Flowrate (m3/s) (meas.) | Flowrate (m3/s) (calc; s+w) | Error (%) |
|---|---|---|---|---|---|---|
| 25.3 | 26.2 | **2.2** | E | 1.9 | 2.53 | -34.9 |
| 26.0 | 26.9 | **2.3** | E | 1.6 | 2.64 | -60.9 |
| 27.8 | 29.0 | **3.6** | E | 1.7 | 3.86 | -123.2 |
| 27.6 | 29.7 | **3.6** | E | 1.8 | 4.09 | -128.8 |
| | | | | | **Avg Error(%)** | **95.8** |

Table 6.4(c) Comparison of measured and calculated ventilation rates (wind direction is E)

| Avg. Temp. Internal (°C) | Avg. Temp. External (°C) | Wind Speed (m/s) | Wind Dir'n | Flowrate (m3/s) (meas.) | Flowrate (m3/s) (calc; s+w) | Error (%) |
|---|---|---|---|---|---|---|
| 19.0 | 11.3 | 3.2 | W | 5.51 | 4.54 | 17.6 |
| 18.8 | 10.9 | 3.1 | W | 6.73 | 5.98 | 11.1 |
| 18.0 | 11.1 | 4.2 | W | 7.08 | 6.80 | 4.0 |
| | | | | | **Avg Error(%)** | **12.2** |

Table 6.4(d) Comparison of measured and calculated ventilation rates (wind direction is W)

It is evident from the above tables, that poor predictions of ventilation rates are achieved for the auditorium when considering the effects of both wind and stack effects, when internal temperatures are close to external temperatures. This is due to the paucity of data on pressure coefficients for non-standard external facade configurations. The above estimates were found with relatively crude data (i.e. coefficients for low rise rectangular blocks).

It is likely that better predictions would be obtained if pressure coefficients had been found from a reduced scale model of a section of the Queens Building and surrounding buildings tested in a wind tunnel.

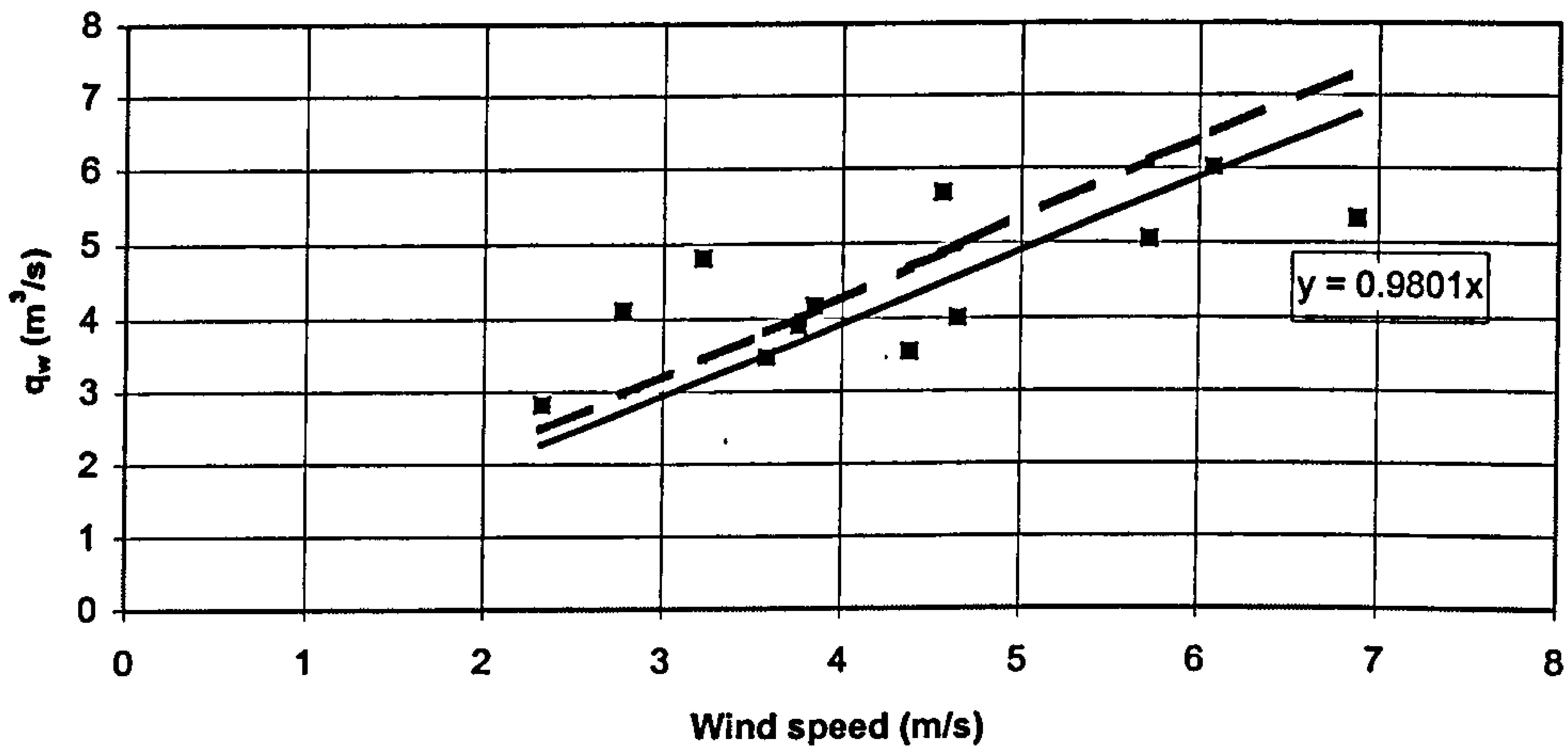



Figure 6.6 Wind speed versus measured and calculated flowrates (dashed line) (second line is "best fit" line)

Data was obtained for flowrates and windspeeds for one particular day. It was assumed that the stack effect was relatively small (i.e. average internal temperature was c. 13 °C; average external temperature was 10 °C) and that flowrates were directly produced via wind induced pressure differences.

Accurate wind speeds and directions were available from instrumentation located "on site" via a weather station located on the roof of Gateway House.

It can be seen from figure 6.6 above, that relatively accurate predictions of flowrates can be produced if more accurate values for wind speeds are known and that a reasonable correlation exists between wind speed and flowrate for this particular set of data. Calculated values shown in figure 6.6 were obtained using the simple ventilation model referred to earlier (see section 6.2.1).

### 6.2.3. Air speeds

To be able to predict thermal comfort conditions in a space, it is important to be able to determine the maximum values of room air speeds, turbulence intensities (and air temperatures). It is assumed that a critical zone occurs adjacent to the inlets grilles where the highest room air speeds (and lowest temperatures) were recorded. A model is developed for finding average air speeds adjacent to inlet grilles within the auditorium. It may initially be satisfactory to find the bulk flow rate entering the space and divide this by the free areas of the inlet grilles directly opposite the external inlets.

Air speeds 0.5 metres from inlet diffusers on both sides of the auditorium have been measured for mid-season and summer external conditions. When these air speeds are plotted against average stack air speeds reasonably good correlations are found (see figures 6.7, 6.8 and 6.9). If an air speed equivalent to the flow rate divided by the supply grille free area (14.7 $m^2$) is also plotted, it indicates that a good approximation of average air speed (in the vicinity of the grilles) can be found from these values. The calculated average speed lies between air speeds found on either side of the space. Hence the suggested empirical "model" gives:

$$v_{inlet} = \frac{q_{stack}}{14.7} \qquad (6.13)$$

where $v_{inlet}$ = average air speed near inlet grilles (m/s);

$q_{stack}$ = stack flow rate ($m^3/s$);

14.7 = Inlet grill free area ($m^2$);

Flow directions can be assumed to be at right angles to the grille face.

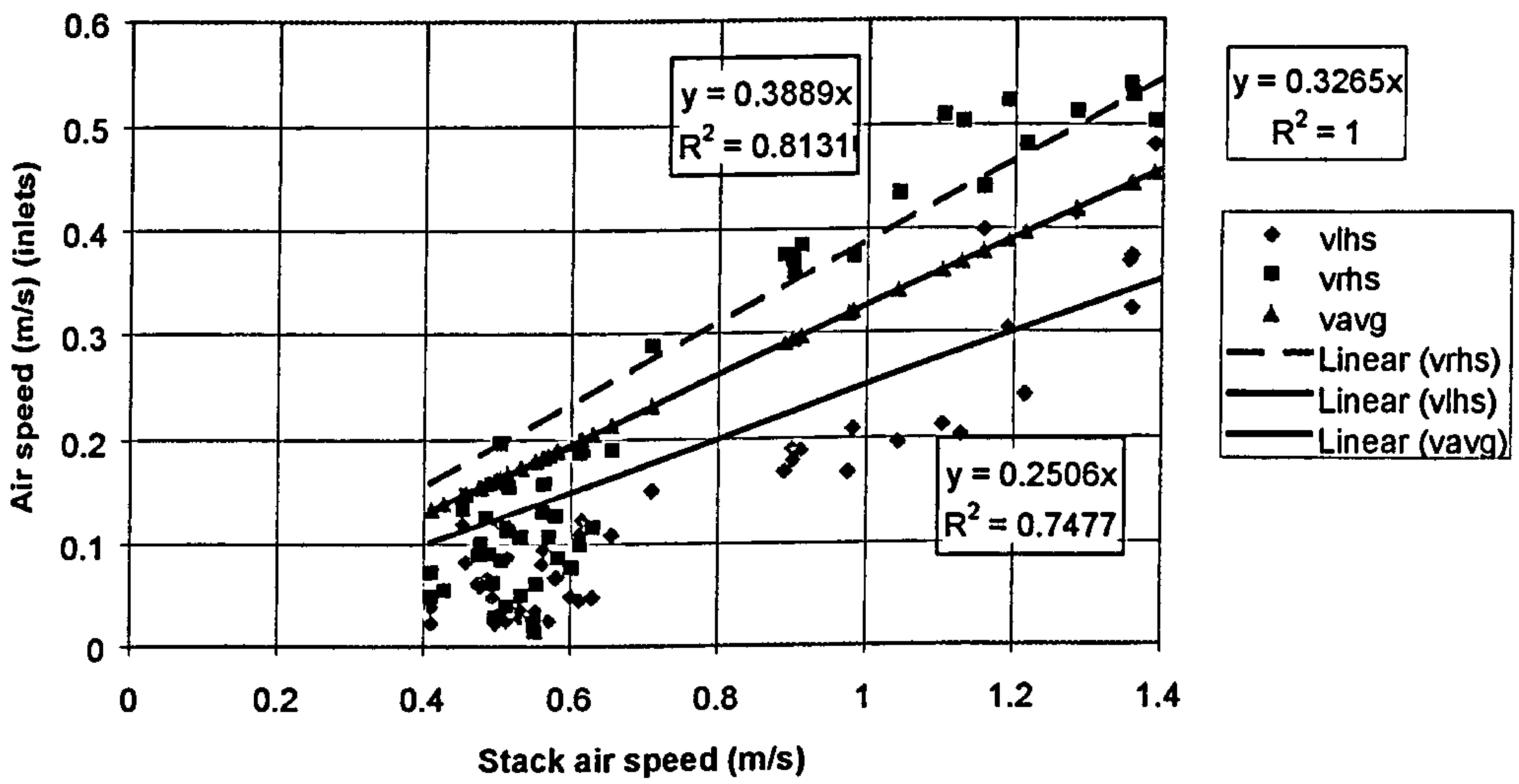



Figure 6.7 Stack air speeds versus inlet air speeds (mid-season external conditions)

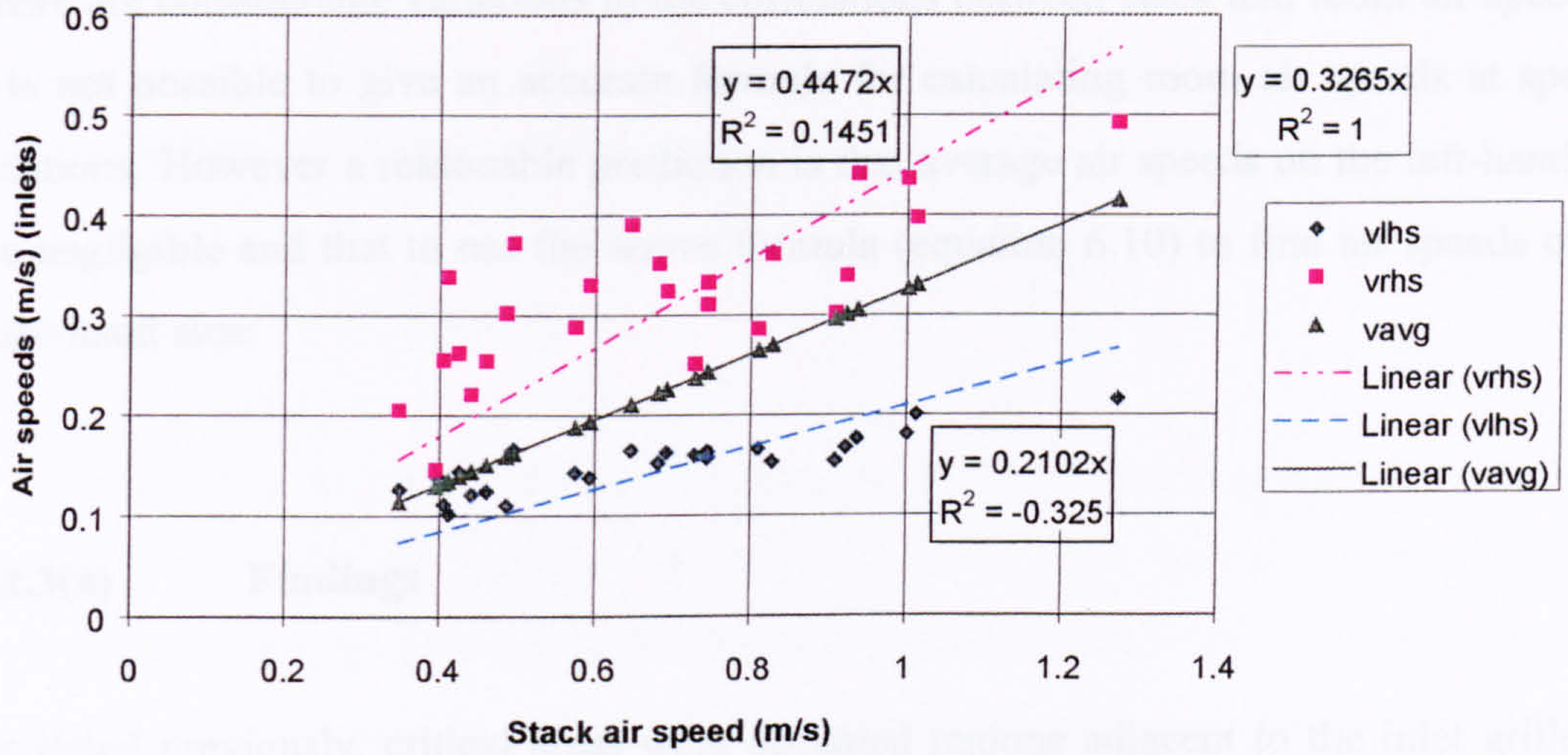



Figure 6.8 Stack air speeds versus inlet air speeds (summer external conditions)

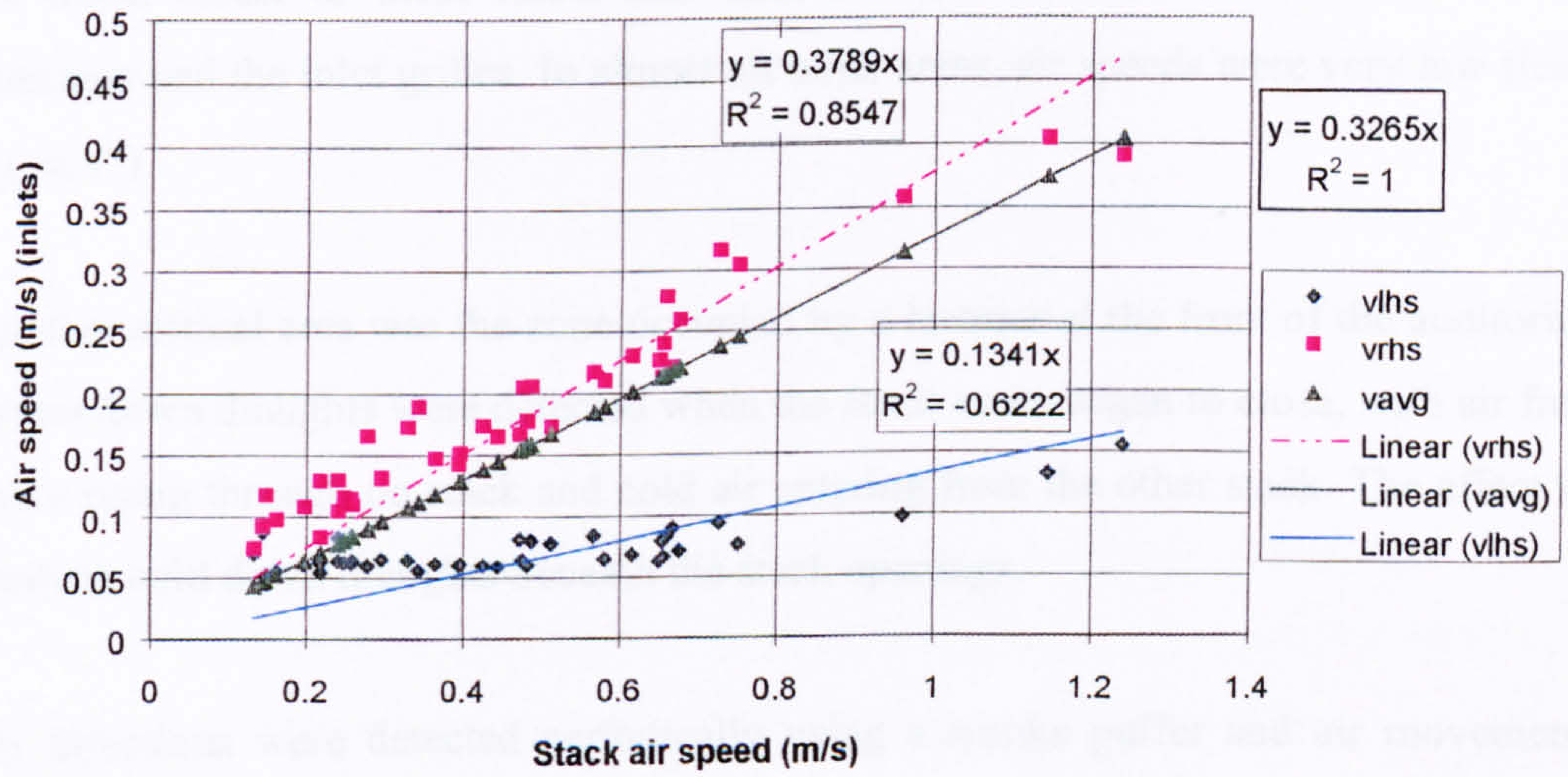



Figure 6.9 Stack air speeds versus inlet air speeds (summer external conditions)

There are considerable variations in the correlations between stack and room air speeds so it is not possible to give an accurate formula for calculating room air speeds at specific locations. However a reasonable prediction is that average air speeds on the left-hand side are negligible and that to use the above formula (equation 6.10) to find air speeds on the right-hand side.

#### 6.2.3(a) Findings

As stated previously, critical areas were occupied regions adjacent to the inlet grilles on the left hand side of the space (at approximately ankle height underneath the seats) where the air speeds were highest and low temperatures (down to 7 °C in winter) were experienced. The left hand side experienced much higher air speeds as the external inlets are much closer to these inlets and there are few obstructions between the external openings and the inlet grilles. In almost all other areas, air speeds were very low (less than 0.1 m $s^{-1}$).

Another critical area was the zone occupied by a lecturer at the front of the auditorium. In winter down draughts were detected when the stack vents began to close, with air from the space rising through on stack and cold air entering from the other stack. The effect was to produce cold down draughts beneath the stack openings.

Air directions were detected periodically using a smoke puffer and air movement was generally from the inlet grilles towards the stack inlets.

Some air turbulence was noted near the ceiling due to the fluctuations in air temperatures in this region.

## 6.3 Prediction of internal temperatures

### 6.3.1 Calculation of time constants

#### 6.3.1(a) Winter conditions

It is desirable to be able to predict internal air and fabric temperatures for a range of external conditions. As can be seen from figure 6.10(a) the air temperature rises asymptotically to the room set point when the heating system is on, then drops once the system has been switched off. It is evident that the start-up internal temperature is dependent on the day of the week (an extended plant shut-down period allows the building to lose more heat and reach a lower internal temperature) and external temperature. In figure 6.10(a) (winter conditions), the internal temperature drops to about 13 °C (Monday morning) and a pre-heat period of c. 4 hours is needed to bring the room temperature up to 19 °C. Temperatures for milder (mid-season) conditions are shown in figure 6.11(a) and the corresponding temperature drops to only 18 °C. In this case, the pre-heat time has dropped to 2 hours. Thus, as the external temperature increases, the pre-heat time reduces. This pre-heat period is a function of external temperature, and the temperature at start-up (of the heating plant) [21].

If the "heating" and "cooling" time constants were known, then it is likely that a first order model of the system can be produced. This, when validated, could be used to make temperature predictions and estimations of heating energy consumption.

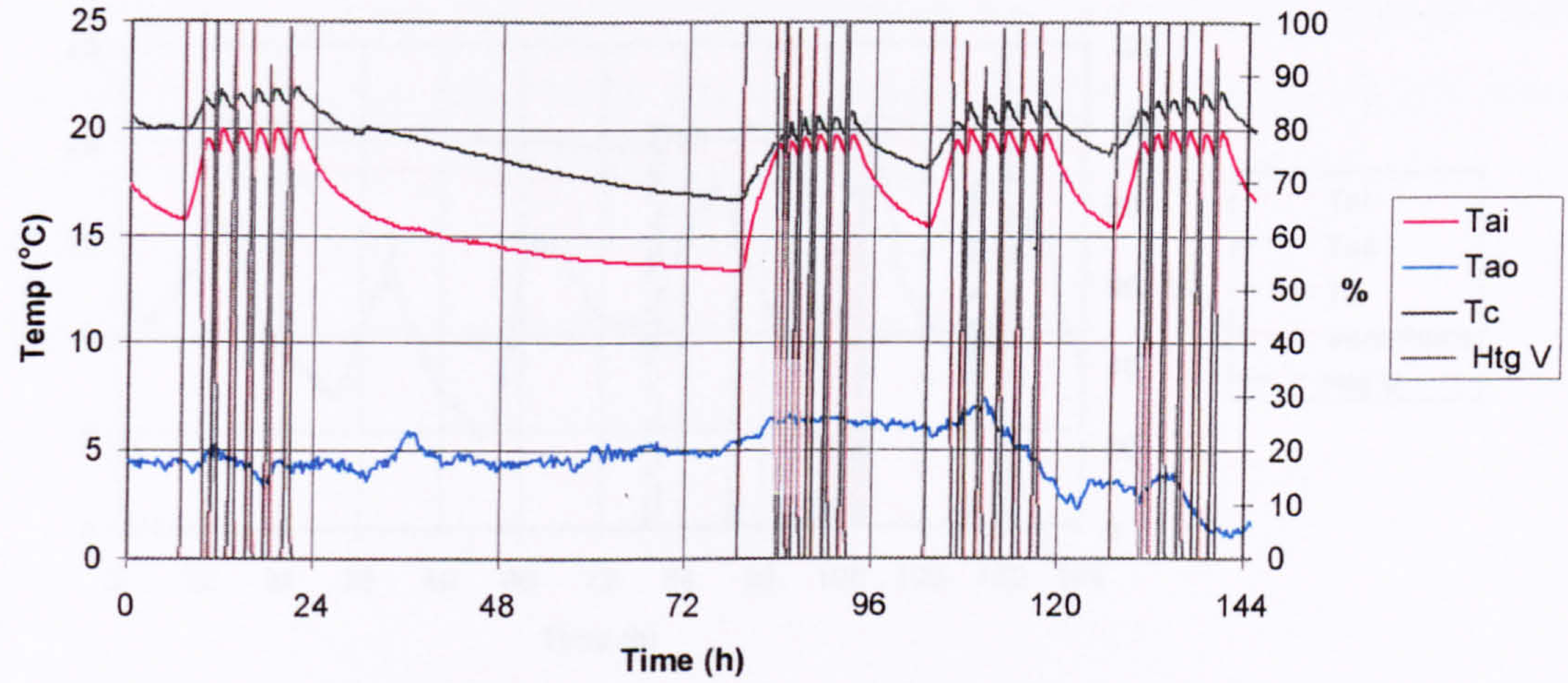



Figure 6.10(a) Internal and ceiling slab temperatures for winter external conditions. The position of a heating valve is also shown.

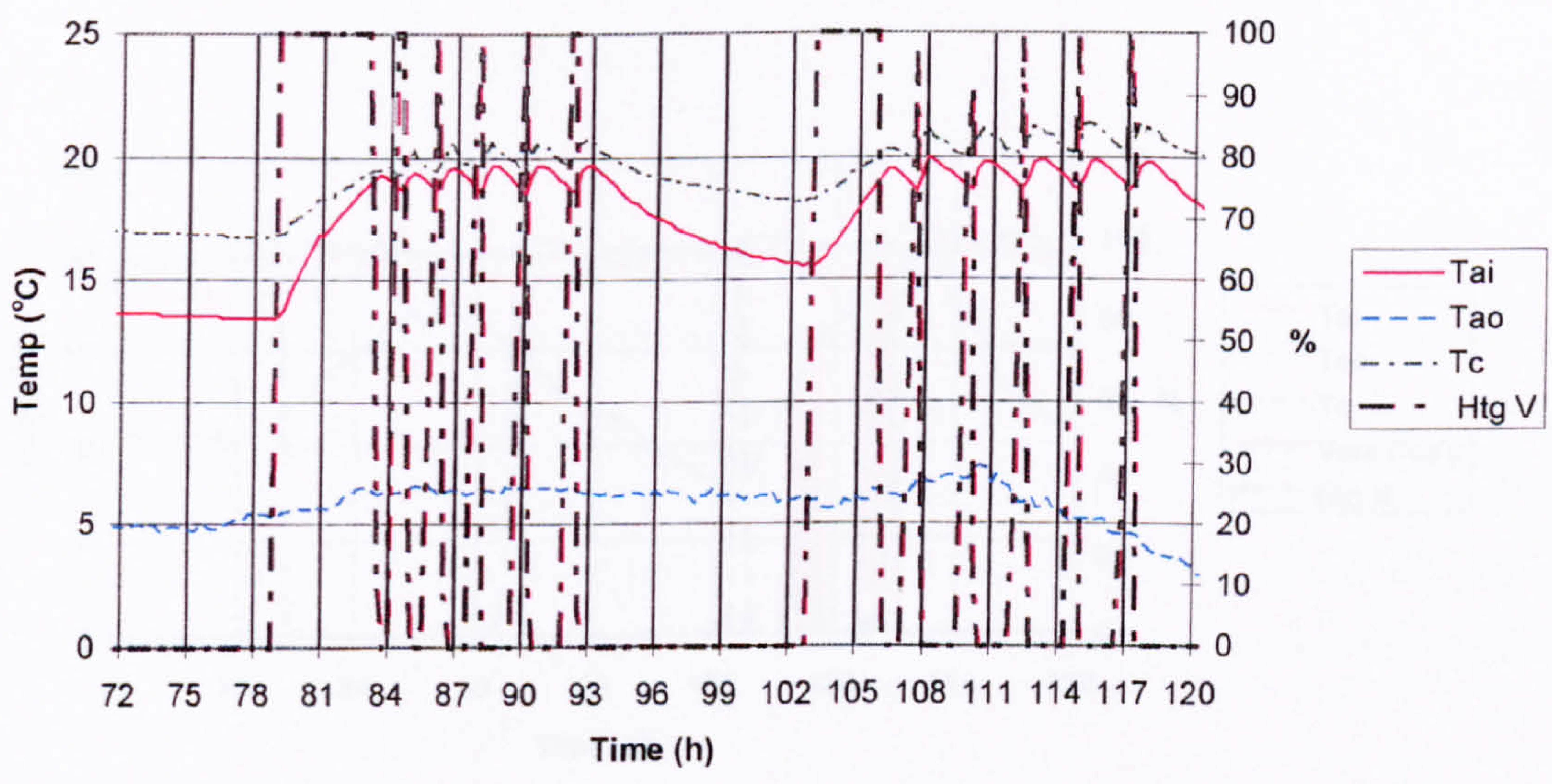



Figure 6.10(b) Internal and ceiling slab temperatures for winter external conditions; two weekdays (Monday and Tuesday) after a weekend heating plant shut-down. The position of a heating valve is also shown.

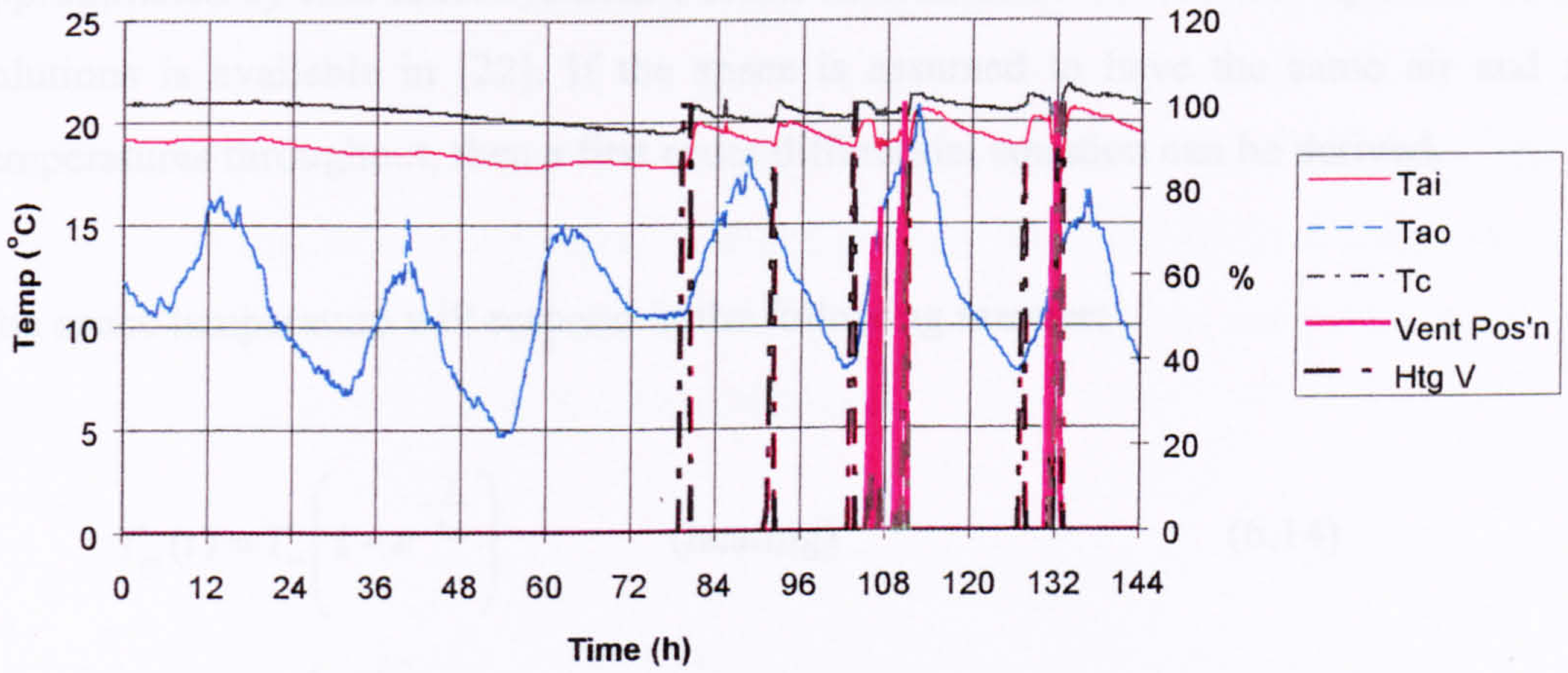



Figure 6.11(a) Internal and ceiling slab temperatures for mid-season external conditions; 72 hours is equivalent to a Monday (after a weekend heating plant shut-down). The position of a heating valve is also shown.

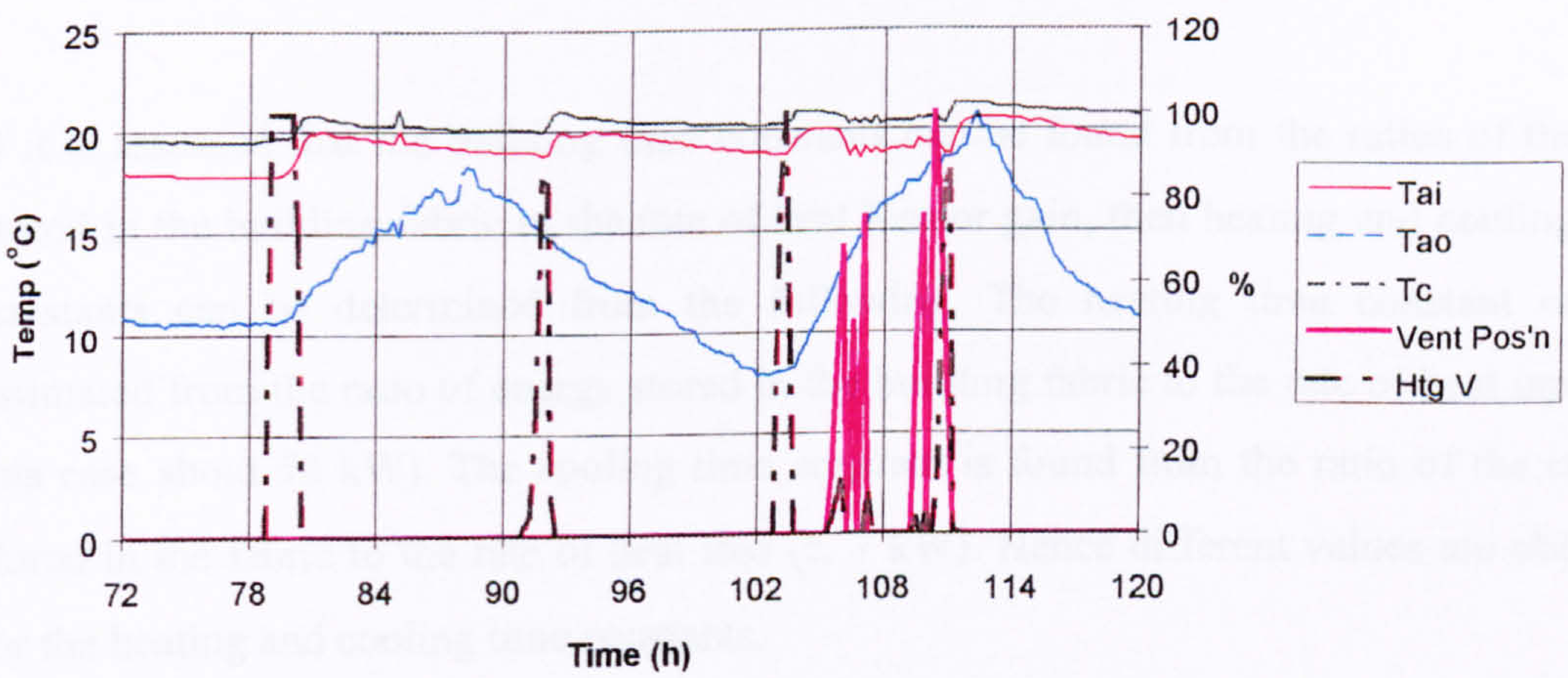



Figure 6.11(b) Internal and ceiling slab temperatures for mid-season external conditions; 72 hours is equivalent to a Monday (after a weekend heating plant shut-down). The position of a heating valve is also shown.

From the above plots, it is evident that the space cooling and heating curves can be approximated by first order systems. Further information on first order equations and their solutions is available in [22]. If the space is assumed to have the same air and fabric temperatures throughout, then a first order differential equation can be derived.

The space temperature will respond in the following manner:

$$T_{ai}(t) = T_{\infty}\left(1 - e^{-\frac{t}{\tau_H}}\right) \qquad \text{(heating)} \qquad (6.14)$$

$$T_{ai}(t) = T_{\infty}\left(e^{-\frac{t}{\tau_C}}\right) \qquad \text{(cooling)} \qquad (6.15)$$

where $T_{\infty}$ = steady state temperature;

$\tau_H$ = heating time constant;

$\tau_C$ = cooling time constant.

If it is assumed that the building time constants can be found from the ratios of the heat stored in the building fabric to the rate of heat loss or gain, then heating and cooling time constants can be determined from the following. The heating time constant can be estimated from the ratio of energy stored in the building fabric to the rate of heat input (in this case about 50 kW). The cooling time constant is found from the ratio of the energy stored in the fabric to the rate of heat loss (c. 7 kW). Hence different values are obtained for the heating and cooling time constants.

A number of data files are available that give internal and external temperatures for a range of climatic conditions. Heating and cooling time constants can be found from plots of internal temperature versus time. Heating time constants (where the space or building is being heated up) can be obtained from plots of the natural log of (1.0-internal temperature) against time. The slope of this curve will be the heating time constant. For cooling time constants (where the building is losing heat) , if the natural log of the temperature is plotted against time, then the slope of the plot will equal the system cooling time constant.

A series of time constants can be found as above. A table of temperatures and time constants in table 6.5. In a similar manner, cooling and heating time constants can be found for internal surface temperatures. Hence comfort conditions within the space can be predicted (assuming air speeds are below 0.1 m/s).

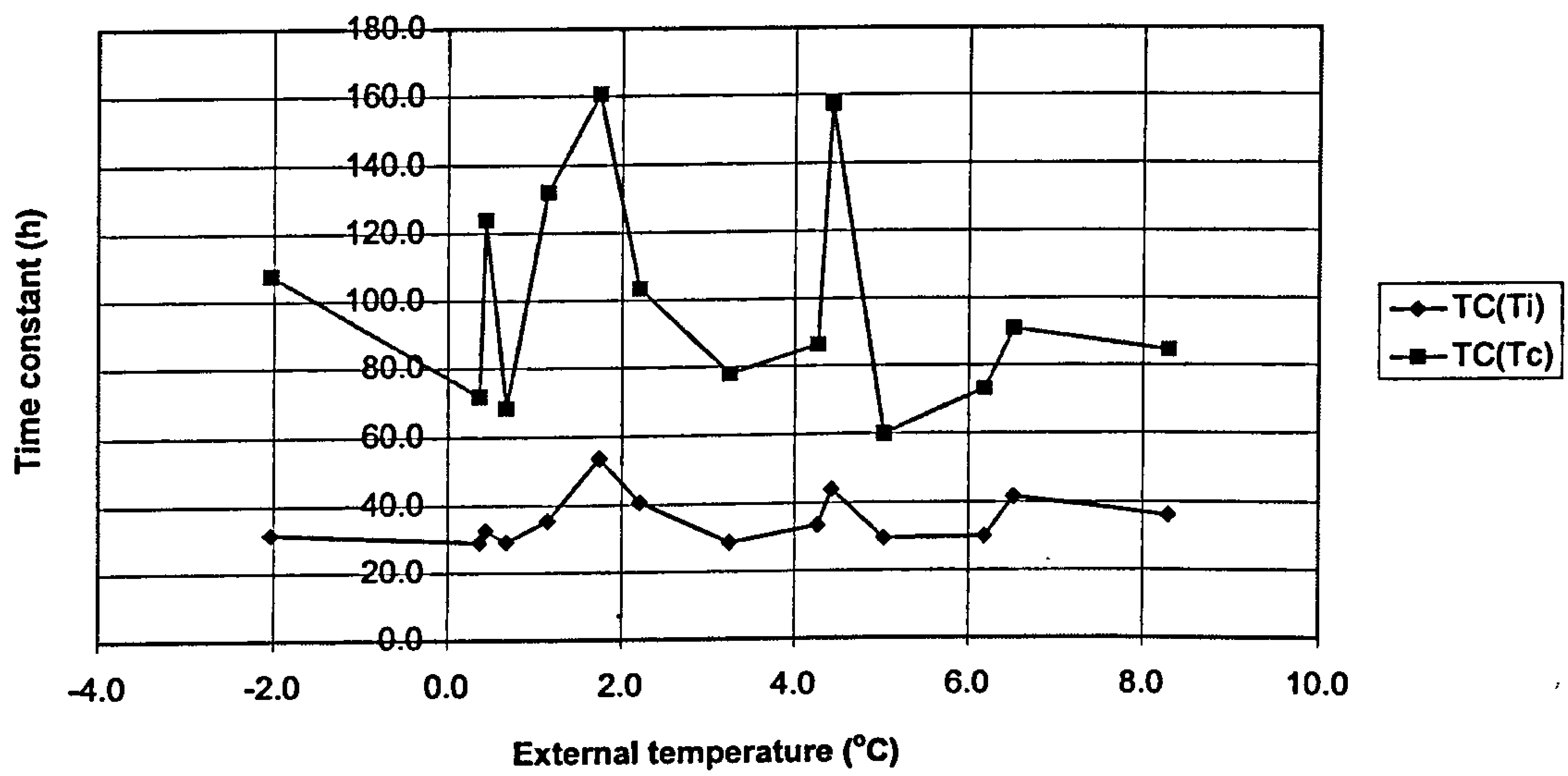



Figure 6.12 Cooling time constants versus external temperature (values calculated from internal air and slab temperatures).

| $t_{ai}$(avg) (°C) | $\tau$(htg) (h) |
|---|---|
| 18.32 | 0.83 |
| 18.34 | 0.95 |
| 18.16 | 1.14 |
| 18.09 | 1.21 |
| 18.09 | 1.24 |
| 17.67 | 1.29 |
| 17.56 | 1.46 |
| 17.57 | 1.54 |
| 16.81 | 2.29 |

Table 6.5 Internal temperatures and room heating time constants (TC)
(large time constants indicate a low rate of heat loss)

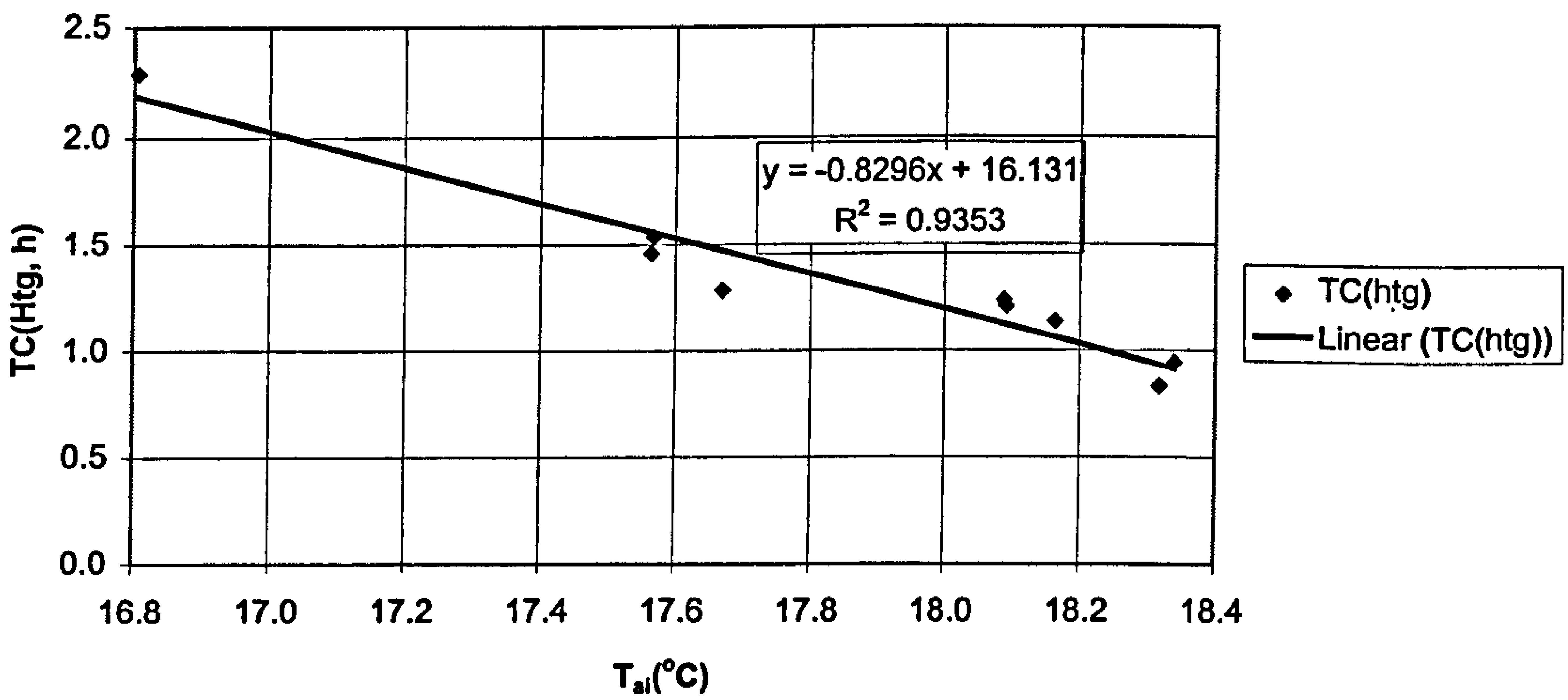



Figure 6.13 Internal temperature versus heating time constant

To generate a model of the behaviour of a building when heating is required over a 24 hour period (or longer), the following variables are needed; i) cooling time constant, ii) heating time constant, iii) external temperature, iv) set point temperature (night set-back), and v) set point temperature (normal operation).

A sample plot of internal temperature can now be found for particular external temperatures and "initial" internal temperatures. Assume the average external temperature is 6.0 °C and the internal temperature at 7:00 is 17.0 °C. The average cooling and heating time constants are found to be 35.5 and 2.0 respectively (see figures 6.12 and 6.13). Again, if the space temperature is taken to be behave like a first order system, then the graph shown in figure 6.14 can be obtained.

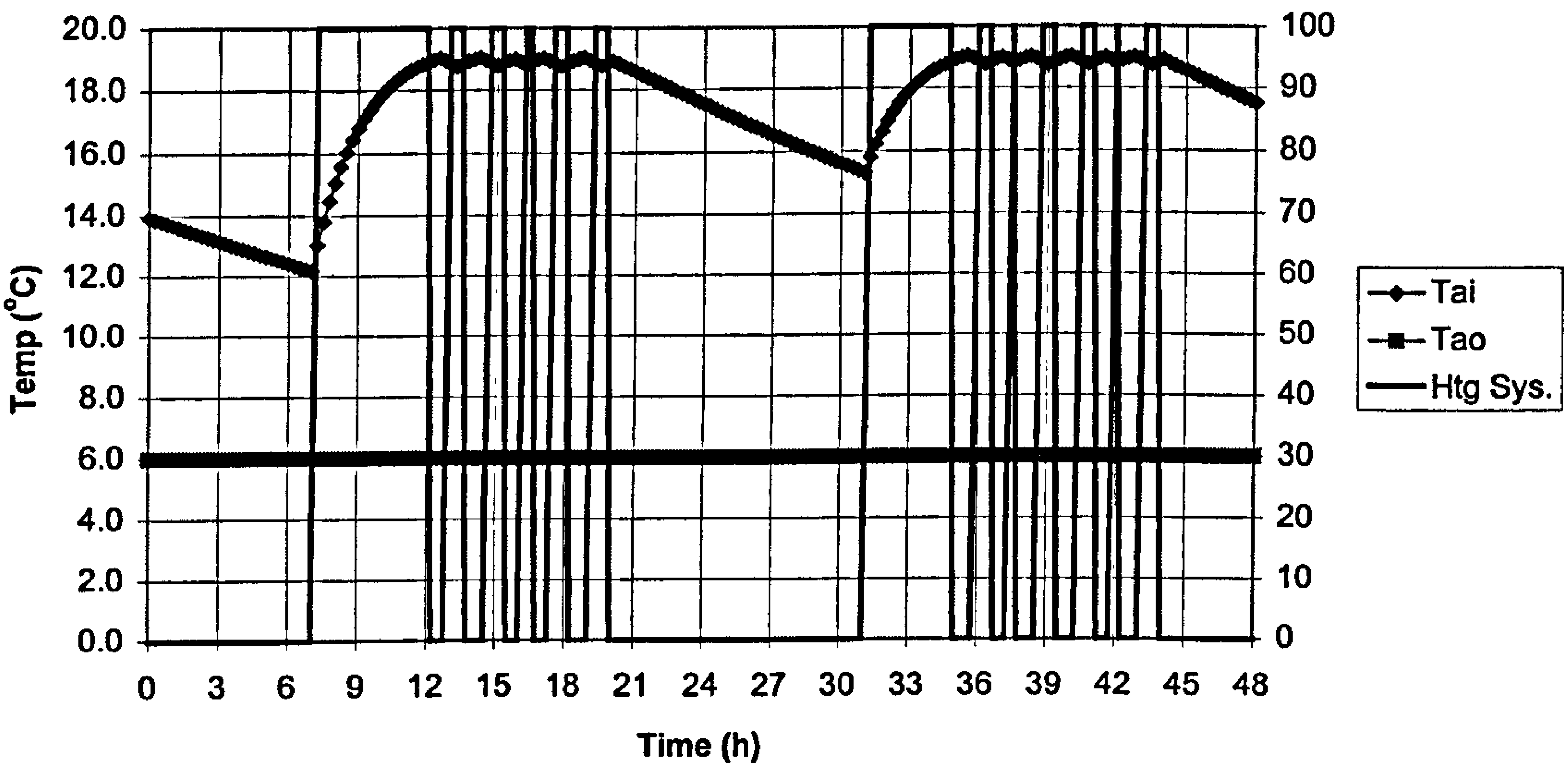



Figure 6.14 Internal temperature and status of heating system for particular cooling and heating time constants (a constant external temperature of 6 °C is assumed).

### 6.3.1(b) Prediction of internal temperatures in winter

Using previously derived time constants and equations 6.11 and 6.12 and knowing the operating times of the auditorium and temperature set points, the internal temperatures can be predicted (see figure 6.15). See appendix G1, which shows a page from the spreadsheet program used to produce figure 6.15.

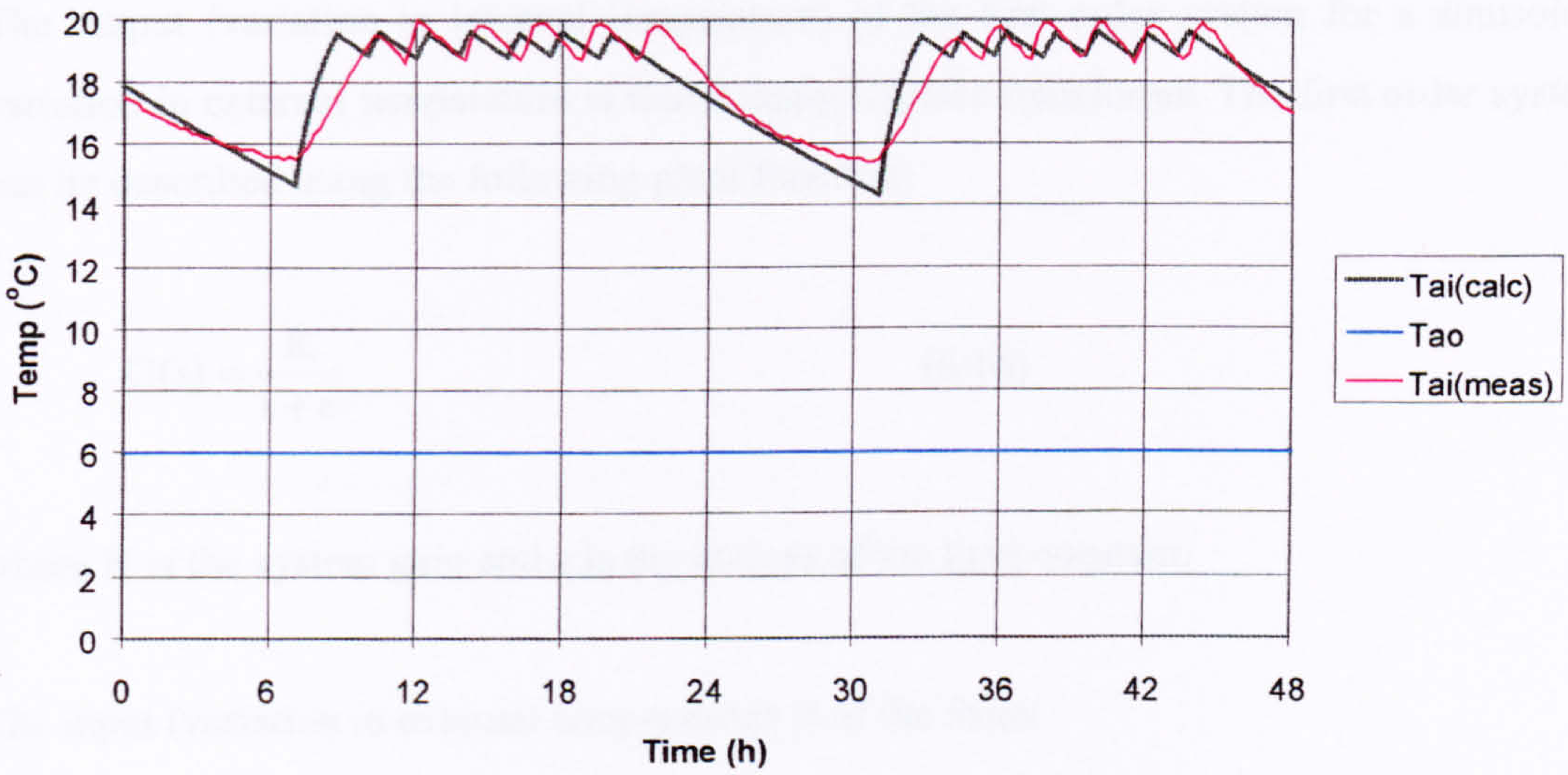



Figure 6.15 Comparison of measured ($T_{ai\ (meas)}$)and predicted internal air temperatures ($T_{ai\ (calc)}$) using a simple first order model.

### 6.3.2 Summer conditions

Internal temperatures in summer can be predicted using a simple model. The behaviour of the model can be calculated assuming it has first order characteristics (using the time constants already derived) and the output will be determined for a sinusoidal variation in external temperature. The variation in time constant with flowrate can be found using the formula supplied in the next section.

The output (variation in internal temperature) of the first order system for a sinusoidal variation in external temperature is found using Laplace transforms. The first order system can be described using the following plant function:

$$G(s) = \frac{K}{s+a} \qquad (6.16)$$

where K is the system gain and a is the inverse of the time constant.

The input (variation in external temperature) is of the form:

$$I(s) = \frac{\omega}{s^2 + \omega^2} \qquad (6.17)$$

where $\omega$ = angular frequency of the external temperature.

Thence the output is:

$$O(s) = G(s)\, I(s) = \frac{K}{s+a}\left(\frac{\omega}{s^2 + \omega^2}\right) \qquad (6.18)$$

The inverse Laplace transform is found; this yields the following coefficients:

$$A1 = \frac{\left[\left(\frac{1}{\tau}\right)^2 - \omega^2\right]}{\left[\left(\frac{1}{\tau}\right)^2 + \omega^2\right]} \tag{6.19}$$

$$A2 = \frac{-k}{2\omega^2} \tag{6.20}$$

$$T_{ai}(t) = T_{base} + T_{var}\left(A1A2 + A2\right)\sin\left((15t + 180)\frac{180}{\pi}\right) \tag{6.21}$$

where $T_{ai}(t)$ = Internal air temperature;

$\tau$ = room time constant (s);

$T_{base}$ = average external temperature (°C);

$T_{var}$ = variation of external temperature

A spreadsheet program (see appendix G2) incorporating the above equation was used to predict internal air temperatures. See the next section to view the outputs of this program.

### 6.3.2(a) Summer conditions - comparison of predictions with experimental results.

Using the formulae above, predictions have been made for internal air temperatures and these are compared with measured values in figures 6.16(a) and 6.16(b).

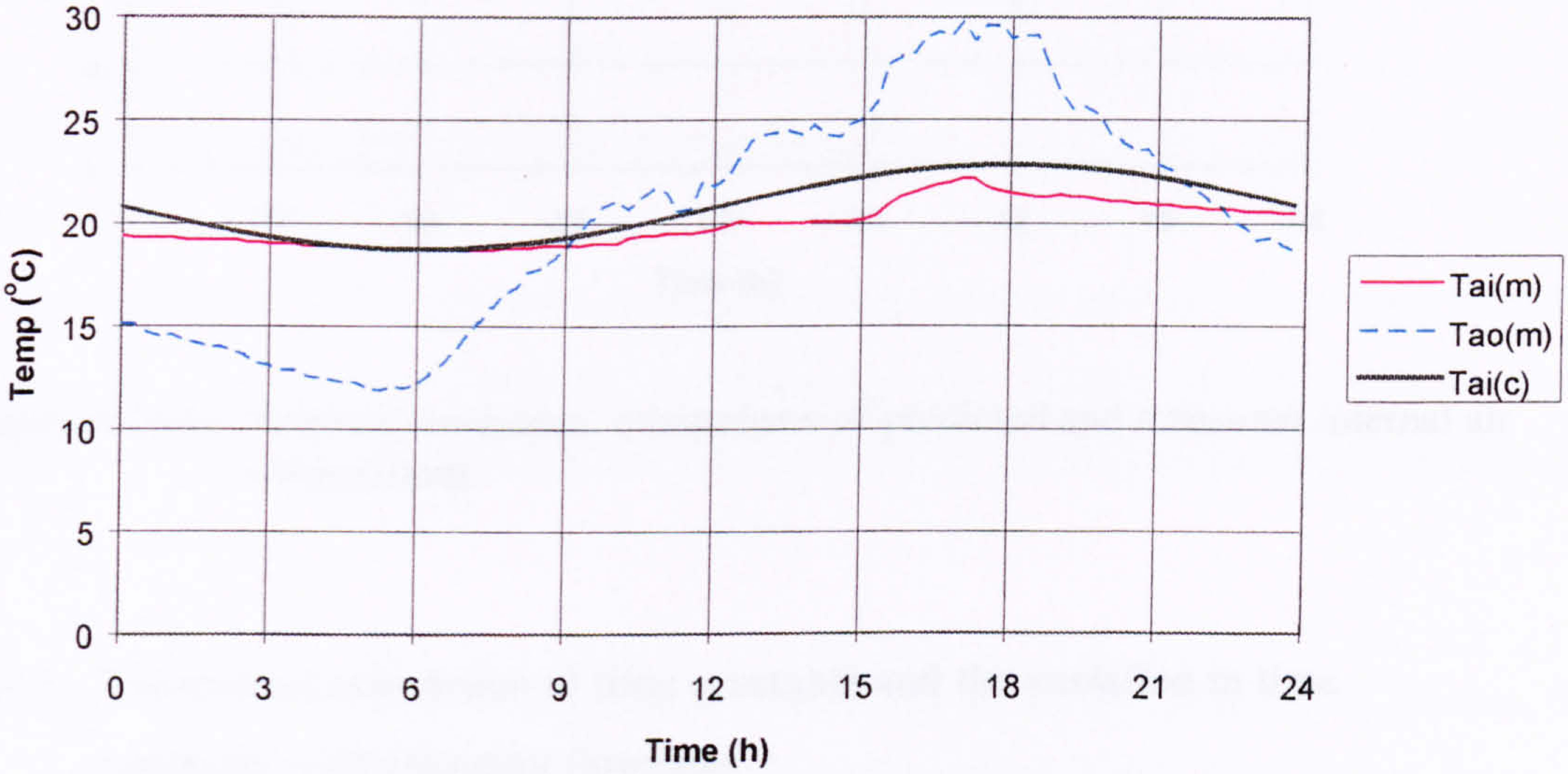



Figure 6.16(a) Summer conditions: comparison of predicted and measured internal air temperatures.

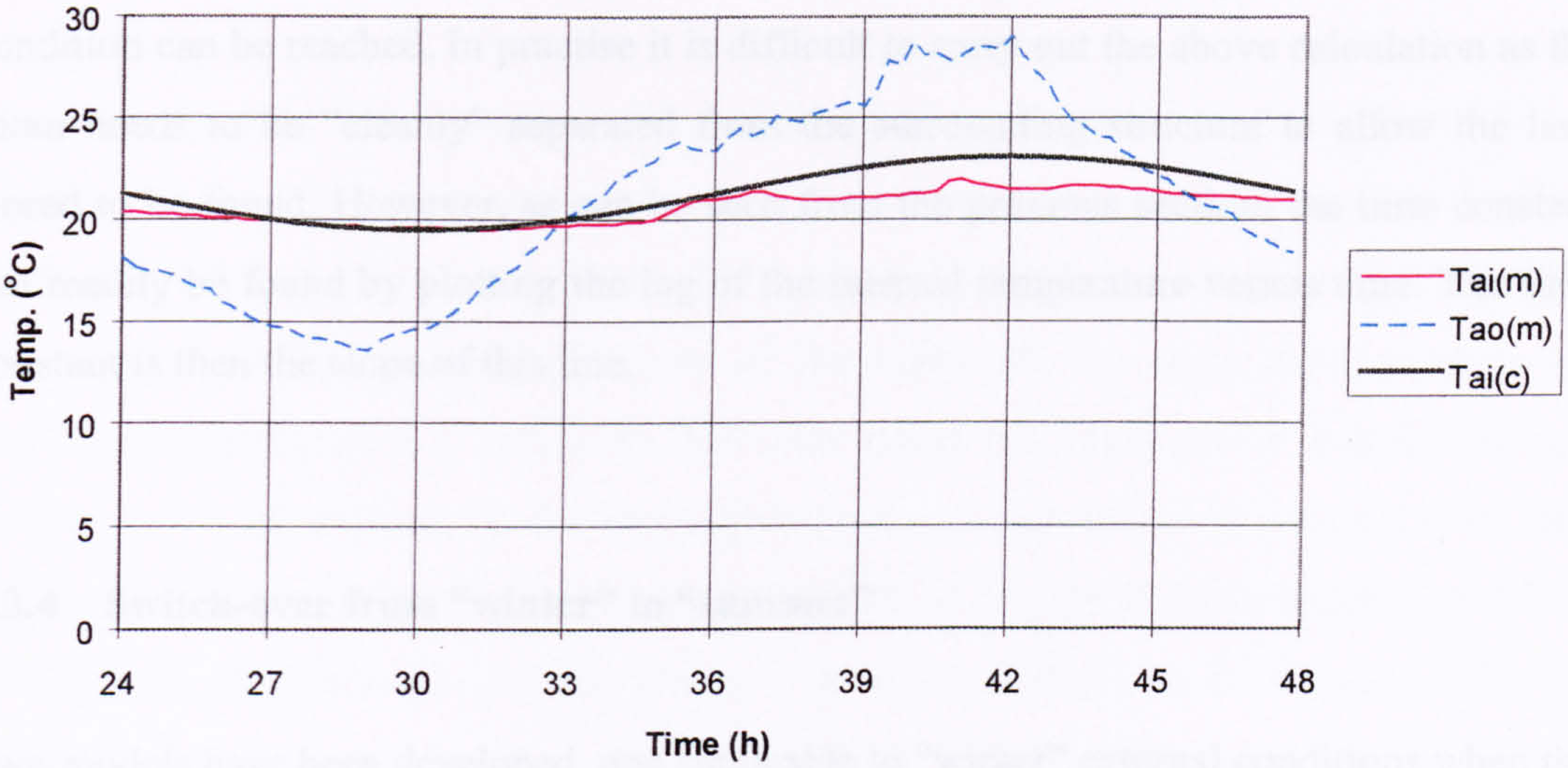



Figure 6.16(b) Summer conditions: comparison of predicted and measured internal air temperatures.

### 6.3.3 Theoretical calculation of time constants and the variation in time constants with changing flowrates

The time constant for a space can be determined by finding the heat stored in the fabric and dividing by the rate of heat loss, i.e.:

$$\tau_f = \frac{\sum q_{mi} C_{pi} (T_{fi} - T_{ao})}{\sum h_i A_i (T_{fi} - T_{ao})} \qquad (6.22)$$

where $\tau_f$ = Fabric time constant (h);

$q_{mi}$ = Fabric component mass (kg);

$C_{pi}$ = Fabric specific heat capacity (kJ kg$^{-1}$ K$^{-1}$ );

$T_{fi}$ = Average fabric component temperature (°C);

$T_{ao}$ = Outside air temperature (°C);

$h_i$ = heat transfer coefficient for surface i (W m$^{-2}$ K$^{-1}$);

$A_i$ = Area of fabric element i (m$^2$);

The above assumes that there are no heat sources in the room and that a steady state condition can be reached. In practise it is difficult to carry out the above calculation as the room needs to be "cleanly" separated from the surrounding structure to allow the heat stored to be found. However, as can be seen from the previous section, the time constant can readily be found by plotting the log of the internal temperature versus time. The time constant is then the slope of this line.

### 6.3.4 Switch-over from "winter" to "summer"

Two models have been developed, one applicable to "winter" external conditions when the predominant effect is produced by the heating system, the other assumes that the main driving effect are the variations in external temperature and to a lesser extent the changes in ventilation rate.

It can be assumed that the "heating model" is used when heating is required, i.e. when the outside temperature is below the room balance point temperature(s). The "summer" model is then used when the outside temperature is above the room balance point temperature and when the heating is off.

### 6.3.5 Effect of time constant on temperature variations

The time constant for a space can be predicted using the formula given above (i.e. in section 6.3.3). It is of interest to know how the time constant will vary with fabric thickness and how the time constant affects internal temperature variations. These have been computed for a rectangular room of approximately the same dimensions as the auditorium (see figure 6.17 and 6.18). The room width and length are 14 m and the height is 6 m. In the calculations the thickness of insulation is assumed to be constant (100 mm) while the thickness of the inner and outer wall layers are varied. It is assumed that heat loss occurs only through one surface (i.e. an external wall) as is the case with the auditorium.

A spreadsheet program (see appendix G3) has been used to generate the following graphs:

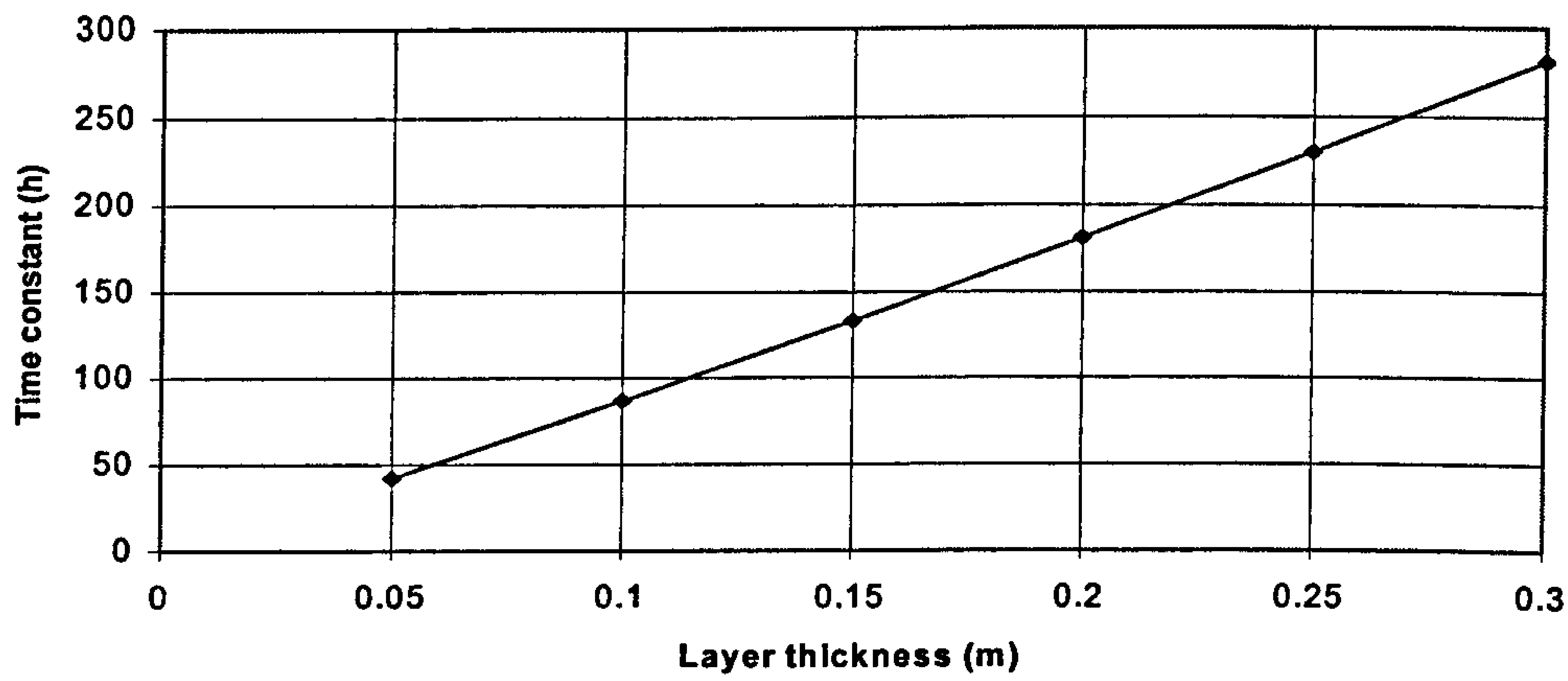



Figure 6.17 Variation of time constant with layer thickness

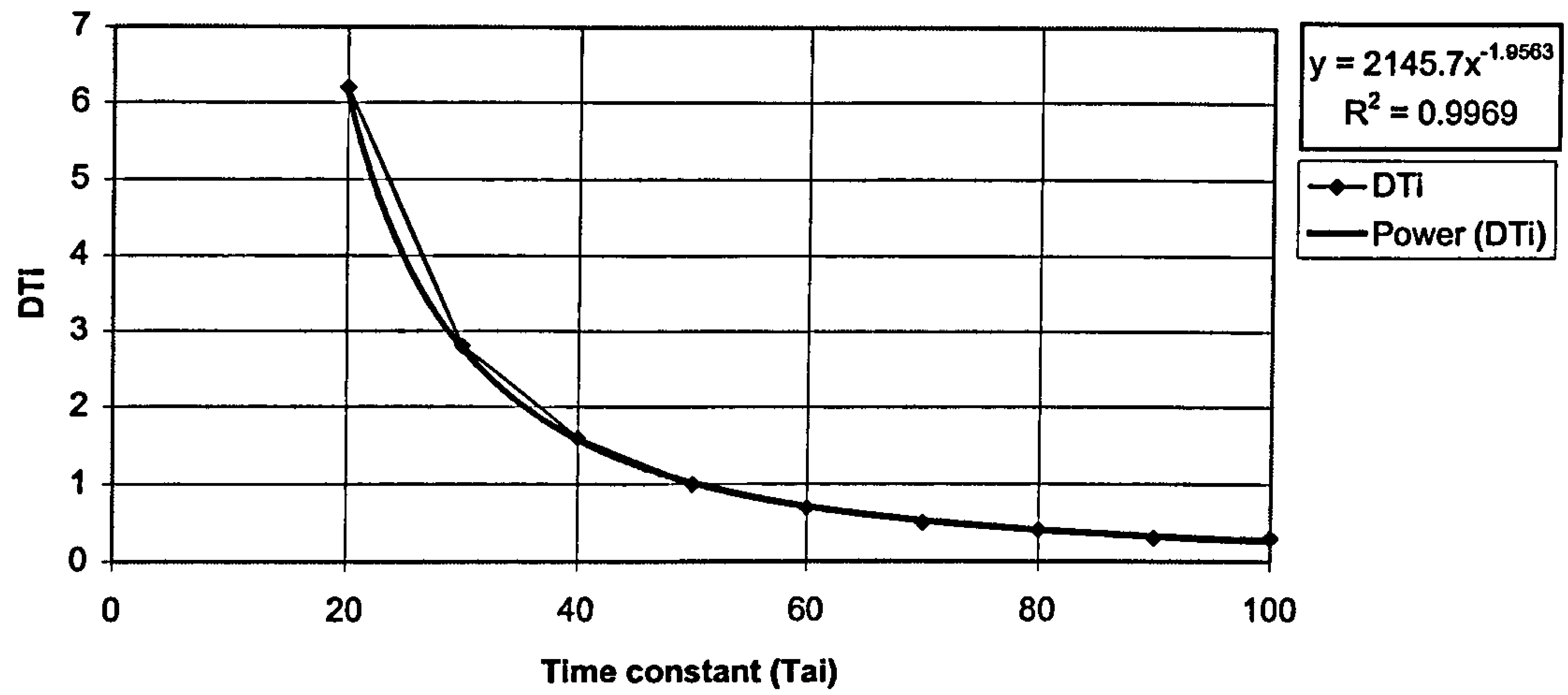



Figure 6.18 Variation in internal air temperature ($DT_i$) with time constant.

## 6.4 Indoor air quality; improving ventilation controls

In winter, ventilation is provided if the internal temperature rises significantly above the set point temperature and/or if $CO_2$ concentrations are above 350 ppm. Assuming that the controls are functioning as designed, this would cause high ventilation areas if the concentration was above 1,000 ppm; and as stated previously, high flowrates would occúr if temperature differences greater than 5 °C were experienced. This would lead to low room temperatures and air speeds above 1.0 m/s.

If the infiltration rate is known and the current $CO_2$ concentration is measured, this can be used to establish the approximate occupancy level:

$$C_i(t_1 - t_0) = (C_i \max - C_0)\left(1 - e^{-N(t_1 - t_0)}\right) \qquad (6.23)$$

$$C_i \max = C_i(t_1 - t_0)\left(1 - e^{-N(t_1 - t_0)}\right) + C_0 \qquad (6.24)$$

Also

$$C_i \max = \frac{F}{q_v} \qquad (6.25)$$

where $F = 5 \times 10^{-6} N_{occ}$

$N_{occ}$ = number of occupants;

$q_v$ = ventilation rate ($m^3/s$);

Hence

$$N_{occ} = q_v C_i \max /(5 \times 10^{-6}) \qquad (6.26)$$

If the number of occupants ($N_{occ}$) is known, the required ventilation rate can be determined; i.e.

$$q_v\ (m^3/s) = 8 \times 10^{-3} N_{occ} \qquad (6.27)$$

The ventilation characteristics have been previously found; the following equation gives a reasonable fit between a function of internal and external temperature and flowrate during the heating season;

$$q_v = 40.0 f_{AE} \left( \frac{T_{ai} - T_{ao}}{T_{ao}} \right)^{0.5} \qquad (6.28)$$

where $f_{AE} = \frac{A_{eff(actual)}}{A_{eff(max)}}$

$A_{eff(actual)}$ = actual effective area ($m^2$)

$A_{eff(max)}$ = Maximum effective area ($m^2$)

$$\text{Thus } f_{AE} = \frac{q_v}{40.0\left(\frac{T_{ai} - T_{ao}}{T_{ao}}\right)^{0.5}} \tag{6.29}$$

Hence the required actual effective area can be determined. The vents setting could be adjusted so that the provided vent area became equal to the calculated area.

## 6.5 Applicability of models

Simple models to allow the prediction of ventilation rates and internal temperatures have been developed. However, these have been produced with limited sets of experimental data, i.e. limited ranges of external temperature, wind speeds and directions and internal heat gains. Ideally more data sets could have been generated, thus extending the applicability of the derived models.

To keep the models relatively simple, a limited number of inputs were used. In practice, ventilation rates and internal temperatures are affected by a range of inputs; these include external temperatures, wind velocities, opening areas, thermal mass of the enclosure, internal gains and other factors.

The applicability of each model will now be considered:

(i) Ventilation model:

This model assumes that buoyancy effects are dominant (i.e. temperature differences are greater than 3 °C and wind speeds are less than 3 m/s).

(ii) "Winter" temperature model:

The variations in internal temperature are assumed to be dominated by the effects of the heating system (i.e. the heat output of the system is much greater than the magnitude of steady-state heat losses or internal heat gains). The system behaves like a first order process.

(iii) "Summer" temperature model:
The changes in internal temperature are considered to be dominated by the effects of external temperature and heat gains are assumed to be low (i.e. at or below average values).

If the above conditions are not satisfied, then the models are not applicable.

## 6.6. Summary

It has been shown that a simple model using temperature differences only can be derived that gives accurate predictions of flowrates if the effective opening area is known.

A simple ventilation model which combines the effects of temperature difference and wind effects can give accurate predictions of flowrate provided accurate values are available for surface pressure coefficients and if the same wind velocities are used in the model and during measurements.

Internal air temperatures in winter can be predicted using a simple first order thermal model if "heating" and "cooling" time constants are available. These time constants are found from experimental data. Internal temperatures in summer can be found using another first order model, assuming a sinusoidal variation in external temperature. Both models have been derived assuming there is one dominating effect driving internal temperatures in the space (the heating system in winter and external temperatures in summer). The effects of ventilation can be modelled assuming increased heat loss from room surfaces, i.e. the time constant varies with the inverse of the ratio of the heat loss occurring when the space is ventilated to the heat loss taking place when the vents are closed. Thus the room time constant reduces when the ventilation rate increases (assuming the external air temperature is less than the internal air temperature).

In summary, then, this chapter has described the following:

- A simple stack effect ventilation model
- A ventilation model which considers the effects of temperature difference and wind effects
- A first order thermal model which can predict internal temperatures in winter
- Another first order thermal model which can determine inside temperatures in summer

# 7 Optimization of the ventilation system

## 7.1 Introduction

In the recent past, building ventilation systems tended to be bespoke systems and the design process had been non-iterative, i.e. one design was produced regardless of whether it was the optimum for a particular building. This was due to time and cost constraints and the unavailability of adequate building simulation tools. However, with the availability of very powerful calculation programs at the engineers disposal this need no longer be the case.

The ventilation of the Queens Building auditorium will now be examined to see if cross-sectional areas and duct sizes can be significantly reduced without compromising the effectiveness of the ventilation system. The results have indicated that the system has been over designed, i.e. vent areas are larger than they need to be; for most of the year excessive flows are generated. This gives rise to high levels of occupant discomfort in winter and mid-season and wastes energy.

The problem of excessive flow rates may be overcome by reducing intake and exhaust vent areas and by improving the control strategy. The system pressure drops can also increased (this can be accomplished by reducing stack cross sectional areas and/or by reducing other opening areas and/or reducing the effective areas of the inlet louvres).

## 7.2 Reducing stack cross-sectional areas

A simple one zone model (see section 6.2.1) can be used to find pressure differences generated by temperature differences and by wind effects. Once available pressure differences have been established, then the flow through the system can be "increased" until the system pressure drops equal those generated by natural forces.

Initially buoyancy induced pressures must be found. If two air masses are separated by a boundary containing two openings (see figure 7.1) then the pressure drops can be determined from:

$$\Delta P_{stk} = (\rho_c - \rho_h) g \Delta h \qquad (7.1)$$

where $\rho_c$ is the density of the outside air

$\rho_h$ is the density of the inside air ,

g is the acceleration due to gravity,

and $\Delta h$ is the difference in height between the centroids of the two openings.

Wind induced pressures are found from:

$$\Delta P_w = 0.5\, \rho\, C_p\, v^2 \qquad (7.2)$$

where $\rho_c$ is the density of the outside air,

Cp is the surface pressure coefficient,

and v is the wind velocity.

Care must be taken with the sign of stack and wind pressures. At each particular point of interest (the location of openings), stack and wind pressures are calculated and added together. The following illustrates the calculation procedure.

The auditorium can be approximated by a single zone containing 3 low level and 8 high level openings.(see figure 7.1). A constant wind speed of 3 m/s at roof height is assumed. The internal and external temperatures are 20 ° and 10 °C respectively.

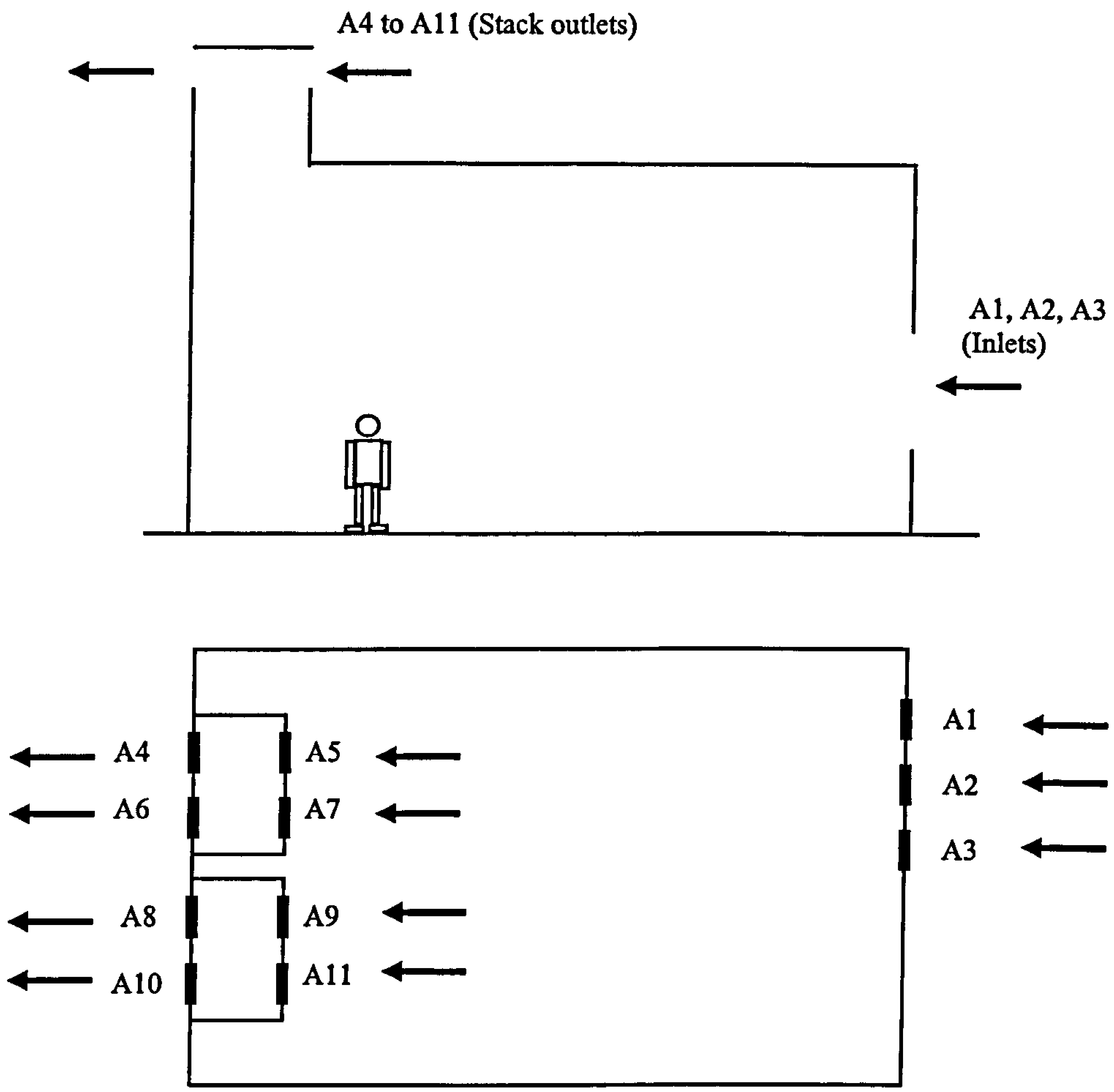



Figure 7.1 Section and plan of auditorium model
A1,A2,A3 low level vents
A4 to A11 high level vents

Calculation of stack pressure drop

$$\Delta P_{stk} = (\rho_c - \rho_h) g \Delta h$$

$$= \frac{P}{R}\left(\frac{1}{T_o} - \frac{1}{T_i}\right) g \Delta h \qquad (7.3)$$

$$\Delta P_{stk} = \frac{1.01x10^5}{278}\left(\frac{1}{283}-\frac{1}{293}\right)9.81 \times 13.7$$

$$= 5.9 \text{ Pa}$$

Stack pressure will be negative at the low level vents and positive at the high level openings (i.e. -5.9 and 5.9 Pa respectively).

Calculation of wind pressures:

$$\Delta P_w = 0.5\ \rho\ C_p\ v^2 \qquad (7.4)$$

An estimate of pressure coefficients is required. Facades facing the wind have a coefficient of 0.5 (i.e. a positive co-efficient) while facades facing the opposite direction have a value of -0.5 (assuming that the stacks can be represented by square sectioned ducts with one edge facing into the wind).

Density is found from:

$$\rho = \frac{P_a}{R_a T_{ao}} \qquad (7.5)$$

$$= \frac{1.01x10^5}{278 \times 283}$$

$$= 1.29 \text{ kg/m}^3$$

Wind pressure is then:

$$\Delta P_w = 0.5\frac{1.01x10^5}{278 \times 283}0.5 \times (3)^2$$

$$= 2.9 \text{ Pa}$$

A table can now be produced listing stack, wind and total pressures at each opening position.

| Opening | Stack pressure (Pa) | Wind pressure (Pa) | Total pressure (Pa) |
|---|---|---|---|
| A1 | 0 | 2.9 | 2.9 |
| A2 | 0 | 2.9 | 2.9 |
| A3 | 0 | 2.9 | 2.9 |
| A4 | -5.9 | -2.9 | -8.8 |
| A5 | -5.9 | 2.9 | -3.0 |
| A6 | -5.9 | -2.9 | -8.8 |
| A7 | -5.9 | 2.9 | -3.0 |
| A8 | -5.9 | -2.9 | -8.8 |
| A9 | -5.9 | 2.9 | -3.0 |
| A10 | -5.9 | -2.9 | -8.8 |
| A11 | -5.9 | 2.9 | -3.0 |

Table 7.1. Stack, wind and total pressures for single zone model

The equation for volumetric flow through a particular opening j is:

$$q_{v_j} = k_j\left(\Delta P_j\right)^{n_j} \tag{7.6}$$

where $k_j$ is the opening coefficient,

$\Delta P_j$ is the pressure drop across the opening and

$n_j$ is the opening exponent (= 0.5 for "large" openings) [18]

The mass flow is then:

$$q_{m_j} = \rho_i q_{v_j} \tag{7.7}$$

where $\rho_i$ = air density

Mass balance must be satisfied, i.e.;

$$\sum_j^m \rho_i q_{v_j} = 0 \tag{7.8}$$

where m = number of openings.

To establish the flow rate for the above example, the internal pressure must first be found. This is obtained from the equality:

$$3k_1(P_1 - P_{int})^{n_1}$$
$$+\left[k_5(P_5 - P_{int})^{n_5} + k_7(P_7 - P_{int})^{n_7} + k_9(P_9 - P_{int})^{n_9} + k_{11}(P_{11} - P_{int})^{n_{11}}\right]$$
$$= \left[k_4(P_4 - P_{int})^{n_4} + k_6(P_6 - P_{int})^{n_6} + k_8(P_8 - P_{int})^{n_8} + k_{10}(P_{10} - P_{int})^{n_{10}}\right] \quad (7.9)$$

Flows are assumed to follow the directions shown in figure 1.3. Flow coefficients are found from:

$$q_{v_j} = k_j(\Delta P_j)^{n_j} \quad (7.10)$$

also

$$q_{v_j} = C_d A_j\left(\frac{2\Delta P_j}{\rho_i}\right)^{n_j} \quad (7.11)$$

Thus

$$k_j = C_d A_j\left(\frac{2}{\rho_i}\right)^{n_j} \quad (7.12)$$

Hence the following opening characteristics can be found (assuming internal air density equals external density):

| Opening | Flow coefficient, $k_j$ (Pa) | Flow exponent, $n_j$ (Pa) |
|---|---|---|
| A1 | 1.5120 | 0.5 |
| A2 | 1.512 | 0.5 |
| A3 | 1.512 | 0.5 |
| A4 | 0.725 | 0.5 |
| A5 | 0.725 | 0.5 |
| A6 | 0.725 | 0.5 |
| A7 | 0.725 | 0.5 |
| A8 | 0.725 | 0.5 |
| A9 | 0.725 | 0.5 |
| A10 | 0.725 | 0.5 |
| A11 | 0.725 | 0.5 |

Table 7.2 Flow coefficients and flow exponents.

The internal pressure is determined by finding a solution to 7.7 by iteration.

The equation then needs to be solved for a range of wind speeds and directions. Solutions for these conditions are shown in tables 7.3(a) and 7.3(b) for two different wind directions.

| $v_w$ (m/s) | DT (°C) | | | | |
|---|---|---|---|---|---|
| | 0 | 5 | 10 | 15 | 20 |
| 1 | 2.17 | 6.30 | 8.79 | 10.77 | 12.48 |
| 2 | 4.34 | 6.99 | 9.36 | 11.25 | 12.93 |
| 3 | 6.50 | 7.56 | 10.05 | 11.94 | 13.56 |
| 4 | 8.66 | 9.22 | 10.68 | 12.63 | 14.25 |
| 5 | 10.87 | 11.33 | 11.63 | 13.23 | 14.94 |

Table 7.3.(a) Flow rates for wind speeds ($v_w$) from 1 to 5 m/s,
temperature differences (DT) of 0 to 20 °C, wind angle = 0°.
(i.e. wind coming from the North)

| $V_w$ | DT (°C) | | | | |
|---|---|---|---|---|---|
| (m/s) | 0 | 5 | 10 | 15 | 20 |
| 1 | 2.17 | 5.61 | 8.31 | 10.38 | 12.15 |
| 2 | 4.32 | 4.47 | 7.29 | 9.66 | 11.58 |
| 3 | 6.52 | 5.00 | 6.47 | 7.71 | 10.29 |
| 4 | 8.68 | 7.76 | 6.94 | 8.24 | 9.03 |
| 5 | 10.84 | 10.16 | 9.24 | 8.56 | 9.75 |

Table 7.3(b) Flow rates for wind speeds from 1 to 5 m/s, temperature differences of 0 to 20 °C, wind angle = 180°. (i.e. wind coming from the South).

Duct pressure loss tables are generated for a range of flow rates. These will establish what the system pressure losses are for the particular flow rates chosen. Total pressure losses can then be compared to the driving pressures that are available.

9 sections in the "duct" system have been assumed. Pressure loss factors for each section are found from the CIBSE Guide section C [23]. Velocities through each section are determined, then velocity pressure losses are found. Table 7.4 details the calculated values for the existing space and table 7.5 gives the total system pressure drops for flow rates from 1 to 10 m3/s. The cross-sectional area of the chimneys is then "reduced" and system pressure drops are recalculated. This procedure may be carried out for a number of other areas; the results of the "75%" and "50%" cases are shown in table 7.5 (i.e. the system pressure drops when the cross-sectional areas have been reduced by 75% and 50% from base values).

| No. | Section | Flow $(m^3/s)$ | Area $(m^2)$ | Vel. (m/s) | Press. Loss Factor ($\xi$) | $\Delta Pl$ (static) | $\Delta Pl$ (vel) | Tot $\Delta Pl$ (Pa) |
|---|---|---|---|---|---|---|---|---|
| 1 | Inlet | 0.33 | 1.95 | 0.17 | 0.7 | | 0.01 | 0.01 |
| | louvres | | | | | | | |
| 2 | Inlet | 0.33 | 1.89 | 0.18 | 0.7 | | 0.01 | 0.01 |
| | dampers | | | | | | | |
| 3 | Inlet | 1.00 | 14.70 | 0.07 | 0.8 | | 0.00 | 0.00 |
| | grilles | | | | | | | |
| 4 | Stack | 0.50 | 1.53 | 0.33 | 0.3 | | 0.02 | 0.02 |
| | inlets | | | | 3.0 | | 0.05 | |
| 5 | Stack | 0.50 | 1.31 | 0.42 | 0.3 | 0.0 | 0.03 | 0.03 |
| | "bend" | | | | | | | |
| 6 | Stack | 0.50 | 1.31 | 0.42 | | 1.1 | | 1.09 |
| | length | | | | | | | |
| 7 | Strut | 0.50 | 1.20 | 0.21 | 0.7 | | 0.02 | 0.02 |
| 8 | Stack | 0.50 | 1.20 | 0.21 | 0.3 | 0.0 | 0.01 | 0.01 |
| | "bend" | | | | | | | |
| 9 | Stack | 0.50 | 1.87 | 0.27 | 1.7 | | 0.07 | 0.07 |
| | outlets | | | | | | | |
| | | | | | | | **Total (Pa)** | **1.27** |

Table 7.4. System pressure drop for the base case
(existing configuration; vents fully open; $q_v = 1.0\ m^3/s$)

| $q_v$ | | A1 | B1 | C1 | C2 |
|---|---|---|---|---|---|
| Aeff | | | $0.75A_{st}$ | $0.50A_{st}$ | |
| $q_v$ | $q_v$ | ΔP | ΔP | ΔP | ΔP |
| ($m^3/s$) | (ac/hr) | (100%) | (100%) | (100%) | (50%) |
| 1 | 3.41 | 1.21 | 1.23 | 1.3 | 3.8 |
| 2 | 6.82 | 1.58 | 1.64 | 1.8 | 11.9 |
| 3 | 10.24 | 2.19 | 2.32 | 2.7 | 25.4 |
| 4 | 13.65 | 3.05 | 3.27 | 3.9 | 44.4 |
| 5 | 17.06 | 4.15 | 4.50 | 5.5 | 68.7 |
| 6 | 20.47 | 5.49 | 6.00 | 7.5 | 98.5 |
| 7 | 23.89 | 7.08 | 7.77 | 9.8 | 133.6 |
| 8 | 27.30 | 8.91 | 9.82 | 12.4 | 174.2 |
| 9 | 30.71 | 10.99 | 12.14 | 15.4 | 220.2 |
| 10 | 34.12 | 13.31 | 14.73 | 18.8 | 271.6 |

Table 7.5. System pressure drops for a range of ventilation flow rates, with vents fully and half open for different stack cross-sectional areas.
($A_{st}$ = stack cross-sectional area ($m^2$))

For this exercise, it is assumed that the ventilation system acts like a constant volume system; i.e. a length of duct plus dampers attached to a constant speed fan. Ventilation rates in winter and mid-season can be taken to be equivalent to the fresh air flow rate (150 people times 8 x $10^{-3}$ $m^3/s$ or 1.2 $m^3/s$) while the summer flow rate will be set such that an average inlet grill velocity of 0.3 m/s is achieved (this equates to 0.3 x 14.7 or 4.4 $m^3$ /s). Hence natural forces should be able to produce pressure drops of 1.3 Pa in winter/mid-season and 3.5 Pa in summer, with the vents fully open (these figures can be obtained by interpolation from the above table). The "required" pressure drops are compared with calculated values to see if these flow rates can be achieved. If the pressure drops obtainable are much greater than required, then the stack cross-sectional areas can be reduced.

In winter, flows are largely driven by the stack effect. Hence, a minimum pressure difference can be derived which will allow the minimum flow rate to be reached. In summer, flow rates are driven by wind effects thus it will be relatively straightforward to find the corresponding pressure difference which would provide a specified flowrate. Thus "minimum" wind speeds can be derived for particular directions.

Winter pressure drop:

The stack pressure drop is calculated from:

$$\Delta P_{stk} = \frac{P}{R}\left(\frac{1}{T_{ao}} - \frac{1}{T_{ai}}\right)gh$$

$$\Delta P_{stk} = \frac{1.01x10^5}{278}\left(\frac{1}{T_{ao}} - \frac{1}{T_{ai}}\right)9.81 \times 13.7$$

$$= 48.98x10^3\left(\frac{1}{T_{ao}} - \frac{1}{T_{ai}}\right)$$

The required pressure drop is 1.3 Pa

$$1.3 = 48.98x10^3\left(\frac{1}{T_{ao}} - \frac{1}{T_{ai}}\right) \qquad \text{(vents fully open)}$$

$T_o = 20$ °C

$T_i = 17.7$ °C.

Hence only a very small temperature drop (2.3 °C) is needed to produce the required pressure difference. If the cross sectional areas of the stacks are reduced to 50 % of the actual values, a pressure drop of 5.4 Pa is required. The corresponding required external temperature is then 10.8 °C (temperature difference equals 9.2 °C).

In summer, pressure drops are found from:

$$\Delta P_w = 0.5\ \rho\ \Delta C_p\ v^2$$
$$= 0.5\ (1.2)(0.5 - (-0.5))v^2$$
$$= 0.6\ v^2$$

$$3.5 = 0.6\ v^2$$
$$v = 2.4\ \text{m/s}$$

With vents half open the required pressure drop increases to 9.5 Pa. The required wind speed is then 4.0 m/s. If the cross-sectional areas of the stacks are reduced by 25 %, the required pressure drop increases to 15.8 Pa. The corresponding wind speed is 5.1 m/s.

## 7.3 Summary

Pressure losses through the ventilation system (inlets, inlet plenum, room and stacks) have been calculated assuming a constant flowrate through the system.

Smaller opening areas could have been used without compromising the performance of the system. One stack could effectively be "shut" down in winter as this would not significantly reduce the effective area but it would be difficult to isolate a stack as there is no louvre between the room and the stack (which could prevent air entering the stack). The system could have operated with one stack only with two inlets connecting it to the auditorium.

It should be possible to improve the systems performance further by streamlining the connections from the room to the two stacks. The highest pressure losses occur at these junctions and in the stacks (where the air speeds are highest). One improvement would be to block off the lower section of each stack (preventing incoming air falling into the stacks). The use of turning vanes (where the air changes direction and rises to the top of the stack), a smoother internal surface (inside the stacks) and more aerodynamic outlets would reduce pressure losses further.

# 8 Conclusions and recommendations

## 8.1 Contribution to knowledge on the performance of low energy ventilation systems as applied to auditoria

The measurement results detailed in Chapter 4 have indicated that thermally comfortable conditions and adequate ventilation rates can be achieved throughout the year for a thermally heavyweight, naturally ventilated auditorium. This has never been demonstrated before for this type of enclosure and ventilation system, due to the uniqueness of the space construction, and the type of ventilation system installed. It has also proved to be difficult to measure ventilation rates in the building due the variability of flowrates, and external conditions (wind speeds, and directions).

The results have shown the enclosure can provide comfortable internal conditions which show little variability despite large variations in external temperatures and high maximum temperatures (above 30 °C) in summer. Comfortable internal environments have also been measured for winter external conditions.

## 8.2 Conclusions relating to the ventilation system as designed

### 8.2.1 Ventilation performance

#### 8.2.1(a) Overall ventilation rates

The series of experimental runs carried out during the three seasons showed that adequate amounts of ventilation air were being provided for the range of climatic conditions experienced. Air change rates greater than 20 were measured in "winter" (with a maximum value of about 36) due to the very high stack induced pressure differences available. High ventilation rates were also generated in "mid-season", ranging from 20 to 27 air-changes per hour. High values were also produced in "summer" but a much bigger range of flowrates occurred (3 to 18 air-changes per hour). Adequate amounts of fresh air were provided even when external temperatures exceeded internal temperatures.

Since flowrate values much higher than those required could be generated (i.e. above 20 air changes per hour compared to 4 air changes per hour which would be the fresh air requirement for a fully occupied auditorium), designing the ventilation system assuming the stack effect only produces a very conservative design (i.e. design flowrates will almost always be reached or exceeded).

#### 8.2.1(b) Winter and mid-season ventilation

The results have indicated that stack driven ventilation predominated in winter and mid-season (temperature differences equal to or greater than 10 °C) with flowrates being relatively steady. More than adequate ventilation rates were provided with air entering via the three "street" vents and exhausting through the stacks.

#### 8.2.1(c) Summer ventilation

Ventilation in summer was driven by stack and wind effects with flows being more unsteady than in winter/mid-season due to the nature of wind driven flowrates. Flowrates occurred even when there was little or no temperature difference between inside and outside or when outside temperatures were greater than those occurring inside. Flow was seen generally to enter via the 3 street vents and exhaust via the stacks even when inside temperatures were less than outside temperatures.

### 8.2.1(d) Night venting

Night venting was carried out to pre-cool the internal fabric of enclosure (in particular the underfloor area). It was difficult to quantify the effects of night venting, but it appeared to prevent the internal surface temperatures from gradually rising during a heat wave (i.e. compare figure 4.19 (night venting taking place) and figure 4.20 (no night venting). It can be seen that little or no rise in internal temperature takes place over a 5 day period when night purging is carried out whereas the internal temperatures slowly rise following days when high external temperatures (c. 28 °C) occur.

The controls appeared to be operating as specified (with respect to internal and external temperatures), with the vents opening when the outside air temperature was less than that inside, outside occupancy hours (see figures 4.9 and 4.19).

### 8.2.1(e) Effects of varying stack outlet configurations

The last part of the results section (section 4.12) indicated that ventilation rates could be increased in summer if leeward outlet vents were held open when windward openings were kept closed. A 40 % increase in flowrate could be achieved if this particular configuration was used. This would be of benefit if there was minimal or no stack effect. It appeared that there was rarely a day when no wind or temperature difference between outside and inside occurred.

Difficulties would arise in practice because of the variability of wind direction and the speed with which the outlets could open (they took about 8 minutes to open fully compared to the 2 minutes required for the inlet vents).

### 8.2.1(f) Local air-change rates

The underfloor system used for discharging air into the lecture theatre proved to be an effective means of ensuring a high average air change rate within the space. More than adequate flows of air were introduced into each of the "zones" examined (i.e. two stack zones, plus right and left hand side low-level and high level zones); however excessive

flowrates were recorded in the right hand side, low level (occupied) zone (see figures 4.22 to 4.24). A section of this particular zone was opposite and in line with external fresh air openings. The type of grille used had a very high free area (64 %); this and the very simple configuration used caused very small pressure drops to be generated across the grille. An effective plenum should have a uniform or near uniform pressure differential between the inside of the plenum and discharge openings. Hence the underfloor void did not act as a satisfactory air supply plenum.

## 8.2.2 Thermal comfort parameters

### 8.2.2(a) Winter and mid-season comfort parameters

The heating system installed proved to be effective in maintaining average air temperatures close to the heating set point temperature (19 °C). during the scheduled occupancy periods (9:00 to 22:00 hours; Monday to Friday) when the space was unventilated. As might be expected, greater temperature fluctuations occurred in "winter" than in "mid-season". When ventilation took place, the average temperature dropped by about 3 to 4 °C; however much greater temperature drops occurred in regions on the right hand side opposite external inlet vents. High stack induced flows were generated; this caused large volumes of relatively untreated (i.e. unheated) air to be directly introduced into the space.

The heating valve which controls the flowrate of water to the finned tubes is controlled from the average room air temperature. Normally the valve would be fully open 2 hours before the start of occupancy (i.e. 7:00 hours) and remain open until the average temperature reached the set point temperature. During occupancy the heating valve would open if the room temperature dipped below the control temperature; however the valve might not open until several minutes after the commencement of ventilation. Hence the heating system might not be operational even when high flowrates of cold outside air are occurring.

The rate of heat transfer from the underfloor heating coils only brought a temperature rise of c. 5 °C (for an external temperature of about 7 °C); clearly a problem exists due to poor heat transfer, and if the heating system isn't continuously charged with hot water and high ventilation rates are needed. A suggested solution is provided in the recommendations section.

High air speeds together with low inlet air temperatures were experienced. Even with moderate external temperatures (c. 10 °C), very uncomfortable conditions were produced (see figure 4.7).

#### 8.2.2(b) Summer comfort parameters

The high effective thermal mass of the enclosure proved to be effective in reducing swings in internal temperatures and in minimising the effects of internal heat gains and changing external temperatures.

The effects of high air speeds was not as detrimental as was the case in winter/mid-season as the air temperatures were close to or higher than external temperatures.

| **Parameter** | **Season** | **Performance:** **Good** √√√ **Fair** √√ **poor** √ |
|---|---|---|
| | | |
| Global ventilation | Winter | √√√ |
| (i.e. average room | Mid-season | √√√ |
| Ventilation rate) | Summer | √√√ |
| | | |
| Air distribution | Winter | √ |
| (variation in local air | Mid-season | √ |
| change rate) | Summer | √√ |
| | | |
| Thermal comfort parameters | Winter | √ |
| | Mid-season | √ |
| | Summer | √√√ |
| | | |
| Indoor air quality | Winter | √√√ |
| ($CO_2$ concentrations) | Mid-season | √√√ |
| | Summer | √√√ |
| | | |

Table 8.1 Overview of system performance

## 8.3 Conclusions relating to the models used.

### 8.3.1 Ventilation models

Accurate predictions of stack driven ventilation rates could be made using the formula previously derived ( 6.6 ). Thus when temperature effects were dominant, the formula gave satisfactory predictions of average ventilation rates. However, inaccuracies increased when stack effects diminished relative to wind induced effects. The large differences between measured and calculated flowrates occurred when stack flowrates were similar in magnitude to wind driven flowrates.

To obtain relatively accurate predictions of ventilation rates driven by both stack and wind effects, it was important to have accurate values for pressure coefficients; air speeds and directions. Predictions improved when more reliable air speeds were available (when comparing measured and calculated ventilation rates).

It is reasonable to expect an accuracy of +/- 10 % when predicting stack flowrates assuming an inside outside temperature difference of the order of 10 °C. For lower temperature differences, to maintain the same level of accuracy, use should be made of the best available values for surface pressure coefficients or use one of the zone ventilation models available in the public domain.

### 8.3.2 Thermal models

When viewing plots of internal air and surface temperature versus time plots for days in winter and mid-season, it appears that it should be possible to predict temperatures using a first order model (the temperatures rise asymptotically and fall exponentially and there are good correlations between "least squares" plots and the actual measured data when the natural log of temperatures is plotted against time).

Average cooling time constants were based on figures for internal air and surface temperatures. Average heating time constants were estimated from the rate at which temperatures increased in the morning after an overnight cooling period.

Time constants should be reduced if the enclosure is ventilated. Ventilation will cause the surface heat transfer coefficients to be increased; for low air speeds this will depend on the surface and air temperatures.

## 8.4 Advantages and disadvantages of the ventilation system as designed

### 8.4.1 Advantages

1) Robust design and construction

A relatively simple system was designed and used. The only moving parts are the inlet and exhaust actuators and a stack mounted "ceiling" fan (which is rarely required). Hence due to the paucity of plant, the cost of maintaining the services should be low relative to mechanically ventilated enclosures.

2) Low plant capital and running costs

Little mechanical plant was needed for climate control (i.e. a low temperature hot water heating system plus natural ventilation system was installed). Thus the services capital costs were low and the running costs are likely to be low. It is very unusual for an auditorium with high heat loads to be naturally ventilated. Less plant noise and pollution is produced than an equivalent mechanically ventilated theatre.

3) High ventilation rates

High ventilation rates were recorded in winter and mid-season and adequate ventilation rates (i.e. greater than the CIBSE fresh air requirement per person) were measured in summer.

4) Complimentary architecture

Arguably the buildings architecture is pleasant and in sympathy with surrounding buildings. It also practically demonstrates the desire of the architect and client to produce a low energy, environmentally "friendly" development.

5) Compact design

The enclosure has a high occupancy density (an occupancy level of 150 people over an area of 190 $m^2$) and did not require space for services plant (such as air handling units, associated ductwork and chillers).

6) Satisfactory acoustics

The spaces acoustics are satisfactory; no amplification equipment is usually required and panelling has been added to "tune" the space without adversely affecting the effective thermal mass of the room.

### 8.4.2 Disadvantages

1) Uneven distribution of fresh air:

Generally, much higher local air-change rates have been measured on the right hand side of the theatre. This is partly due to the layout of the space and the positions of the inlets (i.e. only one half of the end wall (or curved surface) is external).

2) Relatively high air velocities in occupied areas:

High air velocities (greater than 0.5 m/s) in occupied areas (at ankle height, adjacent to inlet grilles), and low air temperatures (less than 10 °C) when inlets are fully open ("winter" external conditions). When outside air temperatures are 10 °C or below, the stack effect generates high inlet velocities via inlet plenums and grilles; the heat transfer rate from the underfloor heating coils is insufficient to prevent low air temperatures when flow rates are high (c. 5.9 $m^3/s$). Much of the incoming air bypasses the coils, which would produce occupant discomfort in areas adjacent to the inlet plenums. However, under normal operation, the openings will only remain open for a relatively short period (until the control conditions are satisfied, see section **Building controls,** page 6).

3) Occurrence of down-flows in stacks:

When stack outlets move to their closed positions down-flows can occur. Discomfort can be produced for a short period if the external temperature is much less than the room temperature (i.e. less than (say) 10 °C with an average internal temperature of 20 °C). This problem has been partially overcome by limiting the travel of the outlet actuators if the low level stack air temperatures are less than 12 °C.

The actuator motors, which provide the power to open and close the outlets, are run in sequence. The consequence of this is that, if the vents are initially open, the left hand side openings will start to close before those on the right hand side, thus causing a temporary higher flow resistance through one stack. This results in downflows in one stack with upflows occurring simultaneously in the other chimney (see figures 4.15 and 4.16).

4) Maintenance of adequate ventilation rates with high occupancy levels.

Further commissioning of control system components is required in order to get the ventilation to operate as designed. At the time the tests were performed, the $CO_2$ sensors were not functioning. Hence high concentrations of $CO_2$ (i.e. values above 1,000 ppm) would not have caused the vents to open.

5) Relatively high dust levels.

Dust accumulation has been observed at the base of the stacks and within the inlet plenums. The reduction in the rate of deposition would be difficult to avoid as filters cannot be used to remove dust since they introduce excessive pressure drops into the system. Air quality problems may occur if such dust is suddenly dispersed into the room in high enough concentrations following a rapid increase in inlet velocities.

6) High ventilation rates

In winter ventilation rates can be excessive, and conditions can become uncomfortable. Finer control of ventilation openings is needed when outside air temperatures are much less than internal temperatures.

7) Unwanted mechanical ventilation

The stack fan appears to switch on under certain conditions (i.e. if the room temperature is 3 °C above the room set point temperature and is greater than the outside temperature). Thus the fan can be switched on at night when the set point is 12 °C. This can act against the night ventilation controls, causing an increase in air temperature within the room (stratification is reduced).

8) Untested conditions

The space has yet to experience high heat gains from actual occupants coupled with high external temperatures. Under those conditions, the $CO_2$ controls would open the vents, thus causing the room temperature to increase with a risk of temperatures above 27 °C being reached. However tests were carried using electric heaters with a total output of 16 kW; these demonstrated the satisfactory operation of the controls (using temperatures only) and that internal temperatures no higher than 25 °C were reached (after several hours). In practice, the space was used intermittently; has only reached a capacity of 100 and is generally unoccupied during the warmest period of the year (August).

9) Unsatisfactory seating ergonomics

The central projector screen is only partially visible from certain seating positions (on the extreme left and right of the back row) and the seat backing are uncomfortable for occupants (i.e. legs cannot be extended beyond the seating boundary).

## 8.5 Conclusions relating to the experimental methods used

1) There are likely to be large uncertainties in the measurements due to relatively small number of points used. Ideally at least two samplers should operate simultaneously for the range of flow rates considered (i.e. 1.2 to 8.8 $m^3/s$; excluding infiltration), thus allowing more gas samples to be obtained over a given time period.

2) Air speed and direction need to be measured at the same time. Air direction in the stacks can be monitored from the velocity probe readings when there is (say) a 10 °C temperature difference between inside and outside. Thus air rising up from the room will be indicated by a temperature close to the recorded room temperature while air entering from the top of the stacks will cause a much lower temperature to be detected by the sensors. Room air speeds are difficult to monitor when the room is occupied.

3) Equipment is intrusive and visible, and considerable amounts of time are required to route and position tubing, sensors and associated wiring. Ensuring safety of personnel can be difficult when installing sensors on high level surfaces.

4) It would be relatively expensive to obtain and site large number of surface temperature sensors. Hence interpolation methods are required to allow calculation of intermediate wall and ceiling temperatures. This could then be verified using spot measurements.

## 8.6 Recommendations

### 8.6.1 Recommendations relating to the ventilation system as designed

1) Wind and temperature effects

At the design stage, the effects of both wind and temperature difference should be considered.

2) Conservative opening areas

Due to the over-estimation of design flow rates, smaller fully open areas could be used in winter and mid-season (i.e. one of the stacks could be shut down during these periods).

3) Control of openings

Finer control of the inlets is required such that smaller openings can be achieved when the external temperature is low. This would improve comfort levels for occupants seated directly opposite external openings.

4) Heater locations

Revised locations for the heaters (currently positioned close to inlet grilles) are required to prevent cold air bypassing the heating surfaces and thus causing discomfort.

5) Symmetrical positioning of openings

The inlets and outlets should be positioned symmetrically about the room centre line. Alternatively it should be possible to vary the areas of inlet grilles. This would ensure a more even distribution of fresh air into the space.

6) Rapid action of opening actuators

Use should be made of inlets and outlet dampers that allow openings to be rapidly opened or closed and which operate simultaneously. This would prevent downdraughts occurring when the openings commence closing.

7) Manual control of openings

It would be useful to have manual control of openings in an accessible position (which would allow ventilation control in the event of failure of the B.M.S.). These could then be overridden after a period of 1 hour.

### 8.6.2 General recommendations relating to stack ventilated thermally heavy-weight buildings.

1. Thermal mass

The effective thermal mass of a building is the most important factor in maintaining constant internal temperatures and ameliorating the effects of variable external temperatures and heat gains.

The required ratio of exposed fabric to total fabric can be determined by simulation (e.g. using ESP or a similar program). Acoustic calculations need to be performed to check the reverberation time of an enclosure. The required reverberation time will depend on how the space will be used.

2. Heat gains

A figure of 40 $W/m^2$ is sometimes quoted as the maximum allowable internal heat gain in naturally ventilated buildings [2]. Greater heat gains (above 100 $W/m^2$) can be tolerated if night venting is carried out and if sufficient areas of fabric are exposed. If the building is used only during academic terms, it is unlikely that the periods of maximum occupancy will coincide with days when the highest external temperatures occur.

Orientation of the building can have a major impact on reducing direct solar gains (windows faced North in the case of the Queens Building auditorium thus minimising incident solar gains).

3. Ventilation prediction

A simple formula can be used to predict stack ventilation rates (see section 6.2.1). Wind driven flowrates are much more difficult to predict and need reliable values for pressure coefficients. Wind tunnel experiments and CFD analyses will allow more accurate predictions of pressure coefficients to be made. However it is possible to make reasonably accurate predictions using pressure coefficients for "standard" shapes.

4. Ventilation controls

Control of opening areas is critical in winter (to avoid cold inlet air temperatures). Trickle ventilation should be used in winter and mid-season. Actual ventilation rates are not so critical in summer (when outside temperatures are similar to inside temperatures). Direct routes (for air flow) from outside to inside should be avoided.

Supply plenums should be carefully designed so that uniform air speeds occur across their outlets (supply diffusers); i.e. a high enough pressure drop needs to occur across the plenum openings.

Ventilation openings should operate in unison and open/close as rapidly as possible. Otherwise there is a risk of cold downdrafts taking place.

5. Heating surfaces

The types and positions of heaters and heat transfer surfaces should be carefully considered to ensure adequate heat transfer between these surfaces and incoming air. The minimum supply temperature should be 18 °C or close to the heating set point temperature (when heating is needed).

Checks could be made on the heaters performance using scale models or CFD. This could ensure supply temperatures were high enough and that inlet air speeds were not excessive (greater than 0.25 m/s).

6. Manual control

It is useful to have manual controls for ventilation openings which can be operated from an accessible position. This could then be over-ridden after an interval of one hour to prevent excessive infiltration loss and to avoid compromising the night venting system used.

7. Design/design development resources

Significant resources (i.e. computing hardware and software, scale modelling techniques and staff expertise in the use and applicability of such methods) are required when carrying out ventilation systems design in naturally ventilated buildings. Expertise in modelling and an appreciation of holistic building design needs to be available at the start of a project.

## 8.7 Recommendations on the use of experimental equipment and techniques

1) Ensure that sufficient and appropriate sensing equipment is inserted when the building is being constructed (as part of a building management system).

   This should include the following:

| Sensor type | Accuracy | Minimum number (per 1,000 m$^3$) |
|---|---|---|
| Air temperature | +/- 0.2 °C | 4 (shape dependent) |
| Surface temperature | +/- 0.1 °C | (depends on ceiling area) |
| Velocity (shielded hot-wire) | +/- 0.05 m/s | (4 per 2.5 m$^2$ (duct cross section)) |
| $CO_2$ (Infra-red etc.) | +/- 5 ppm | 1 plus tubing array and local pump per duct. |
| Infra-red (occupancy) | +/- 2 % of full scale reading | 1 per door |

   (Note that velocity probes and tracer gas techniques are used for flow measurement)

2) Carry out periodic recalibrations of sensors and associated equipment (hence there should be a routine procedure for gaining access to high-level areas; the methods used may coincide with those used by the building maintenance personnel).

3) Isolate the area(s) under examination so that uncontrollable influences are minimised (i.e. lock access doors) such that ventilation areas are fully under the operators control.

## 8.8 Recommendations for further research

1) Examination of the effect of varying the amount of (acoustic) cladding on the room air and fabric time constants and the variations in internal temperatures.

   Acoustic cladding needs to be added to room surfaces to reduce the reverberation time and to make sounds more audible. However they detract from the thermal performance of the space and make it more thermally lightweight. It would be useful to examine the effects of varying the area of cladding on temperature variations for different space volumes to see what area is appropriate for a given volume

2) CFD modelling of wind driven ventilation and external flow field profiles

   A major challenge for CFD is the accurate modelling of external flows around buildings. Very large numbers of cells are required and additional equations need to be solved. Accurate modelling would allow wind driven flowrates to be calculated for particular spaces using particular external profiles, varying wind speeds and directions. This would include the calculation of external pressure coefficients for non-standard facades or openings. Effects of adjacent buildings on local wind speeds, directions, flow fields could also be simulated.

3) Accurate simulation of internal air flow using CFD

   It is still difficult to model boundary layers using a relatively coarse grid using CFD. It is necessary to model boundary layer activity accurately if reliable values for heat transfer coefficients are to be obtained. Accurate values at the boundaries would improve the overall accuracy of a simulation as these values are used to determine velocities, temperatures etc. in adjacent cells.

4) Adapting existing dynamic thermal modelling programs to enable modelling of controllable openings.

The program used to simulate the Queens building auditorium had its opening areas set to fixed values throughout the simulation runs. This was unrealistic as the opening areas vary with the conditions; if the space is unoccupied the openings close and the areas used partially depend on occupancy levels. A program should allow such areas to vary in a realistic manner during a simulation

5) Simulation and optimisation of night cooling; improving night cooling controls

   A key to the successful performance of the auditorium in summer (particularly during the heat wave during August 1995) was the fact that night cooling was used to purge the space at night and remove heat absorbed by the internal fabric during the day. The control of night venting needs to be optimised for a given space such that the space is not over or under cooled.

6) Methods for improving heat transfer rates between floor slabs and air flowing over the slabs

   As stated above, night venting was instrumental in producing acceptable temperatures in the theatre during the day in summer. It is important to facilitate the transfer of heat from the floor slab to the air passing over its surface. CFD simulations coupled with dynamic thermal models could be used to optimise floor sizes and shapes for given flowrates.

7) Accurate modelling of floor plenums

   The experimental results showed the lack of symmetry in air speeds and flowrates on both sides of the auditorium. The performance of the plenum could have been improved if it had been modelling using CFD. Various adjustments in inlet areas and the areas used for the supply grilles could have been made to give a relatively uniform distribution of air into the room.

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# Appendices

# Appendix A: Results

## Appendix A1 Winter results

| Run | Vent Settings (%) | Heat Load (OC) (kW) | Air change rates (1/h) | | | | | | Avg ACR (1/h) | Stack | | Avg. Temp. | | | Wind Speed (m/s) | Wind Dir'n |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | LHS st | RHS st | LHS R | RHS R | LHS hl | RHS hl | | Velocity (m/s) | ACR (1/h) | Internal (°C) | External (°C) | Ceil (°C) | | |
| W1 | 50 | 5 | 12.2 | 12.3 | 21.3 | 28.6 | 15.6 | 12.1 | 17.0 | 1.36 | 22.3 | 21.2 | 3.3 | 20.2 | L | NW |
| | 75 | 5 | 33.4 | 38.2 | 28.0 | 40.0 | 32.0 | 21.4 | 32.0 | 1.72 | 28.2 | 20.5 | 3.2 | 19.5 | L | NW |
| | 100 | 5 | 37.2 | 42.1 | 38.9 | 49.9 | 30.8 | 20.0 | 36.5 | 1.8 | 29.5 | 19.8 | 4.4 | 19.3 | L | NW |
| W2 | 50 | 5 | 23.6 | 22.9 | 28.8 | 32.9 | 18.0 | 19.5 | 24.3 | 1.34 | 21.9 | 19.5 | 2.2 | 19.8 | L | WNW |
| | 75 | 5 | 30.8 | 29.7 | 41.9 | 51.9 | 30.8 | 25.9 | 35.2 | 1.77 | 29.0 | 19.7 | 4.3 | 19.1 | L | WNW |
| | 100 | 5 | 22.8 | 22.7 | 34.7 | 84.5 | 27.0 | 21.0 | 35.5 | 1.72 | 28.2 | 19.5 | 4.7 | 19.1 | L | WNW |
| W3 | 50 | 5 | 23.4 | 18.4 | 16.7 | 23.1 | 13.1 | 19.2 | 19.0 | 1.25 | 20.5 | 17.3 | 4.9 | | L | SSE |
| | 75 | 5 | 29.0 | 29.6 | 37.1 | 79.4 | 38.1 | 23.0 | 39.4 | 1.51 | 24.7 | 16.9 | 5.0 | | L | SSE |
| | Infilt. | 5 | 1.1 | 1.2 | 1.1 | 1.1 | 1.1 | 1.1 | 1.1 | | | 18.2 | 5.2 | | L | SSE |
| W4 | Infilt. | var | 1.2 | 1.3 | 0.9 | 1.1 | 1.1 | 1.2 | 1.1 | | | 20.7 | 5.7 | | L | WNW |
| | | var | 0.5 | 0.4 | 0.3 | 0.4 | 0.5 | 0.4 | 0.4 | | | 20.3 | 5.5 | | L | WNW |

Table A1 winter results

## Appendix A2 Mid-season results

| Run | Vent Settings (%) | Heat Load (OC) (kW) | Air change rates (1/h) | | | | | | Avg ACR (1/h) | Stack | | Avg. Temp. | | | Wind Speed (m/s) | Wind Dir'n |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | LHS st | RHS st | LHS R | RHS R | LHS hl | RHS hl | | Velocity (m/s) | ACR (1/h) | Internal (°C) | External (°C) | Ceil. (°C) | | |
| M1 | Infilt. | var | | | | | | | | | | | | | | |
| | | var | 0.3 | 0.2 | 0.2 | 0.2 | 0.2 | 0.2 | 0.2 | | | 19.8 | 9.3 | 19.5 | M | W |
| | | var | 0.1 | 0.1 | 0.1 | 0.1 | 0.1 | 0.1 | 0.1 | | | 20.2 | 9.5 | 19.5 | M | W |
| M2 | 50 | var | 9.9 | 10.3 | 22.2 | 21.3 | 8.8 | | 14.5 | | | 20.2 | 8.0 | 20.5 | L | SSE |
| | Infilt. | var | 0.3 | 0.3 | 0.3 | 0.3 | 0.3 | | 0.3 | | | 21.5 | 7.4 | 21.2 | M | SE |
| M3 | 50 | 5 | 21.0 | 22.3 | 24.2 | 26.1 | 7.6 | 11.1 | 18.7 | 1.09 | 17.9 | 19.9 | 10.0 | | M | WSW |
| | 75 | 5 | 17.0 | 20.0 | 24.0 | 22.5 | 16.9 | 13.8 | 19.0 | 1.26 | 20.7 | 19.2 | 9.2 | | M | WSW |
| | 100 | 5 | 16.0 | 17.6 | 20.5 | 48.1 | 17.4 | 12.8 | 22.1 | 1.39 | 22.7 | 18.8 | 9.2 | | M | WSW |
| M4 | 50 | 5 | 15.1 | 17.6 | 22.5 | 43.6 | 11.2 | 9.8 | 20.0 | 0.93 | 15.3 | 20.4 | 12.0 | 20.9 | L | SW |
| | 75 | 5 | 20.9 | 20.6 | 26.8 | 44.7 | 30.2 | 18.8 | 27.0 | 1.35 | 22.1 | 20.8 | 11.6 | 21.2 | L | SW |
| | 100 | 5 | 22.7 | 18.8 | 18.9 | 19.2 | 17.3 | 20.3 | 19.5 | 1.33 | 21.8 | 20.4 | 11.6 | 20.6 | L | SW |
| M5 | 50 | 5 | 20.5 | 22.4 | 25.4 | 36.4 | 13.6 | 7.2 | 20.9 | 1.15 | 18.8 | 19.0 | 11.3 | 20.0 | L | W |
| | 75 | 5 | 17.2 | 22.1 | 25.9 | 70.1 | 21.2 | 14.6 | 28.5 | 1.40 | 23.0 | 18.8 | 10.9 | 20.0 | L | W |
| | 100 | 5 | 19.0 | 10.9 | 27.2 | 45.8 | 24.4 | 15.6 | 23.8 | 1.48 | 24.2 | 18.0 | 11.1 | 19.9 | L | W |

Table A2(a) mid-season results

## Appendix A2 Mid-season results

| Run | Vent Settings (%) | Heat Load (OC) (kW) | Air change rates (1/h) | | | | | | Avg ACR (1/h) | Stack | | Avg. Temp. | | | Wind Speed (m/s) | Wind Dir'n |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | LHS st | RHS st | LHS R | RHS R | LHS hl | RHS hl | | Velocity (m/s) | ACR (1/h) | Internal (°C) | External (°C) | Ceil. (°C) | | |
| M6 | 0 | Var | 0.5 | 0.3 | 0.6 | 0.6 | 0.5 | | 0.5 | | | 21.3 | 9.1 | 21.1 | L | S |
| M7 | 0 | Var | 0.2 | 0.2 | 0.2 | 0.3 | 0.2 | | 0.2 | | | 20.9 | 9.9 | 20.7 | L | W |
| M8 | 0 | Var | 0.4 | 0.4 | 0.4 | 0.5 | 0.4 | 0.5 | 0.5 | | | 20.5 | 3.6 | | M | W |
| M9 | 0 | Var | 0.8 | 0.8 | 0.8 | 0.8 | 0.9 | 0.8 | 0.8 | | | 21.1 | 14.1 | 20.9 | L | SW |
| M10 | 50 | 5 | 19.5 | 19.9 | 22.6 | 45.4 | 28.2 | 16.3 | 27.1 | | | 20.2 | 14.5 | 20.3 | L | WSW |
| | 75 | 5 | 16.2 | 17.4 | 17.3 | 20.2 | 12.7 | 9.5 | 16.8 | | | 20.1 | 15.3 | 20.3 | L | WSW |
| | 100 | 5 | 23.3 | 22.4 | 19.5 | 39.0 | 43.3 | 14.7 | 27.0 | | | 20.0 | 15.4 | 20.5 | L | WSW |
| M11 | 50 | 5 | 14.4 | 13.6 | 13.6 | 13.9 | 14.0 | 13.8 | 13.9 | | | 20.4 | 10.1 | 20.3 | L | N |
| | 75 | 5 | 6.7 | 6.3 | 7.0 | 8.6 | 6.4 | 6.4 | 7.0 | | | 19.5 | 10.1 | 20.6 | L | N |
| | 100 | 5 | 9.8 | 9.3 | 9.5 | 9.3 | 8.6 | 8.4 | 9.3 | | | 19.6 | 9.8 | 19.9 | L | N |

Table A2(b) mid-season results

## Appendix A3 Summer results

| Run | Vent Settings (%) | Heat Load (OC) (kW) | Air change rates (1/h) | | | | | | Avg ACR (1/h) | Stack | | Avg. Temp. | | | Wind Speed (m/s) | Wind Dir'n |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | LHS st | RHS st | LHS R | RHS R | LHS hl | RHS hl | | Velocity (m/s) | ACR (1/h) | Internal (°C) | External (°C) | Ceil (°C) | | |
| S1 | 50 | 5 | 15.7 | 16.1 | 18.7 | 32.9 | 18.1 | 7.9 | 18.3 | | | 20.2 | 16.8 | 20.5 | L | WSW |
| | 75 | 5 | 16.8 | 14.7 | 13.9 | 29.4 | 28.6 | 6.8 | 18.4 | | | 20.3 | 17.4 | 20.6 | L | WSW |
| | 100 | 5 | 16.4 | 17.1 | 20.1 | 27.2 | 20.0 | 6.9 | 18.0 | | | 20.4 | 17.4 | 20.5 | L | WSW |
| S2 | Infilt. | 0 | 0.2 | 0.1 | 0.1 | 0.1 | 0.1 | 0.1 | 0.1 | | | 21.6 | 25.5 | | L | SSW |
| | 50 | 0 | 1.9 | 1.8 | 1.4 | 1.5 | 1.5 | 3.4 | 1.9 | | | 21.4 | 22.8 | | L | SSW |
| S3 | 100 | 0 | 2.8 | 2.9 | 1.0 | 3.4 | 2.2 | 2.3 | 2.7 | | | 22.2 | 24.7 | 21.8 | L | SSW |
| S4 | 100 | 5 | 5.0 | 5.3 | 6.1 | 7.9 | 4.5 | 4.4 | 5.5 | | | 24.6 | 24.9 | 25.8 | | ENE |
| | 100 | 5 | 8.2 | 7.9 | 9.1 | 9.9 | 7.0 | 5.2 | 7.9 | | | 25.0 | 25.9 | 25.8 | | ENE |
| | 100 | 5 | 5.4 | 7.2 | 2.9 | 2.6 | 3.0 | 3.8 | 4.1 | | | 27.5 | 29.8 | 26.4 | | ENE |
| S5 | 100 | 5 | 6.2 | 6.8 | 6.4 | 7.0 | 6.7 | 5.1 | 6.4 | | | 25.3 | 26.2 | 26.2 | | E |
| | 100 | 5 | 5.3 | 5.4 | 5.6 | 6.0 | 5.2 | 6.1 | 5.6 | | | 26.0 | 26.9 | 26.2 | | E |
| | 100 | 5 | 8.0 | 9.0 | 4.5 | 3.8 | 4.5 | 5.8 | 5.9 | | | 27.8 | 29.0 | 26.6 | | E |
| | 100 | 5 | 8.9 | 6.4 | 6.2 | 6.8 | 4.9 | 3.5 | 6.1 | | | 27.6 | 29.7 | 26.6 | | E |

Table A3(a) summer results

## Appendix A3 Summer results

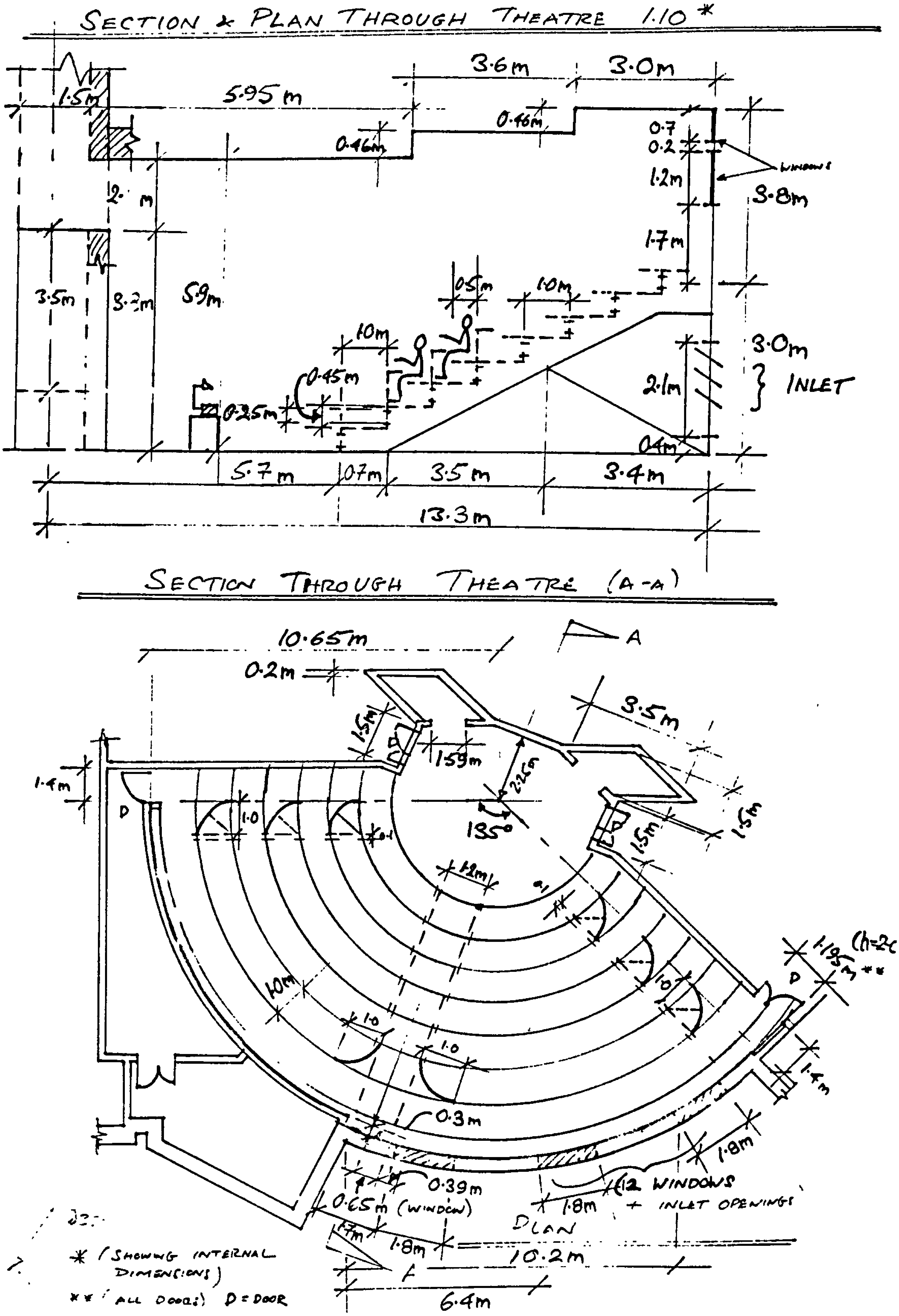
| Run | Vent Settings (%) | Heat Load (OC) (kW) | Air change rates (1/h) | | | | | | Avg ACR (1/h) | Stack | | Avg. Temp. | | | Wind Speed (m/s) | Wind Dir'n |
|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|---|
| | | | LHS st | RHS st | LHS R | RHS R | LHS hl | RHS hl | | Velocity (m/s) | ACR (1/h) | Internal (°C) | External (°C) | Ceil (°C) | | |
| S6 | 50 | 5 | 11.4 | 12.8 | 17.3 | 30.4 | 12.9 | 9.5 | 15.7 | | | 23.4 | 21.1 | 23.8 | | NE |
| | 100 | 5 | 10.3 | 10.9 | 13.2 | 18.1 | 11.0 | 8.8 | 12.0 | | | 23.5 | 21.6 | 23.8 | | NE |
| | 100 | 5 | 13.0 | 14.8 | 24.1 | 18.5 | 16.0 | 7.7 | 15.7 | | | 22.8 | 21.7 | 23.6 | | ENE |
| S7 | Infilt. | 5 | 1.2 | 1.1 | 1.1 | 1.2 | 1.1 | 1.2 | 1.2 | | | 23.8 | 23.1 | 23.4 | | E |
| | 50 | 5 | 11.7 | 7.1 | 17.2 | 28.6 | 23.9 | 12.2 | 16.8 | | | 23.6 | 17.8 | 23.5 | | SSE |
| S8 | Infilt. | 5 | 0.4 | 0.4 | 0.7 | 0.4 | 0.3 | 0.3 | 0.4 | | | 24.3 | 25.6 | 24.3 | | E |
| | Infilt. | 5 | 0.2 | 0.2 | 0.4 | 0.3 | 0.3 | 0.3 | 0.3 | | | 25.8 | 28.5 | 24.9 | | E |

Table A3(b) summer results

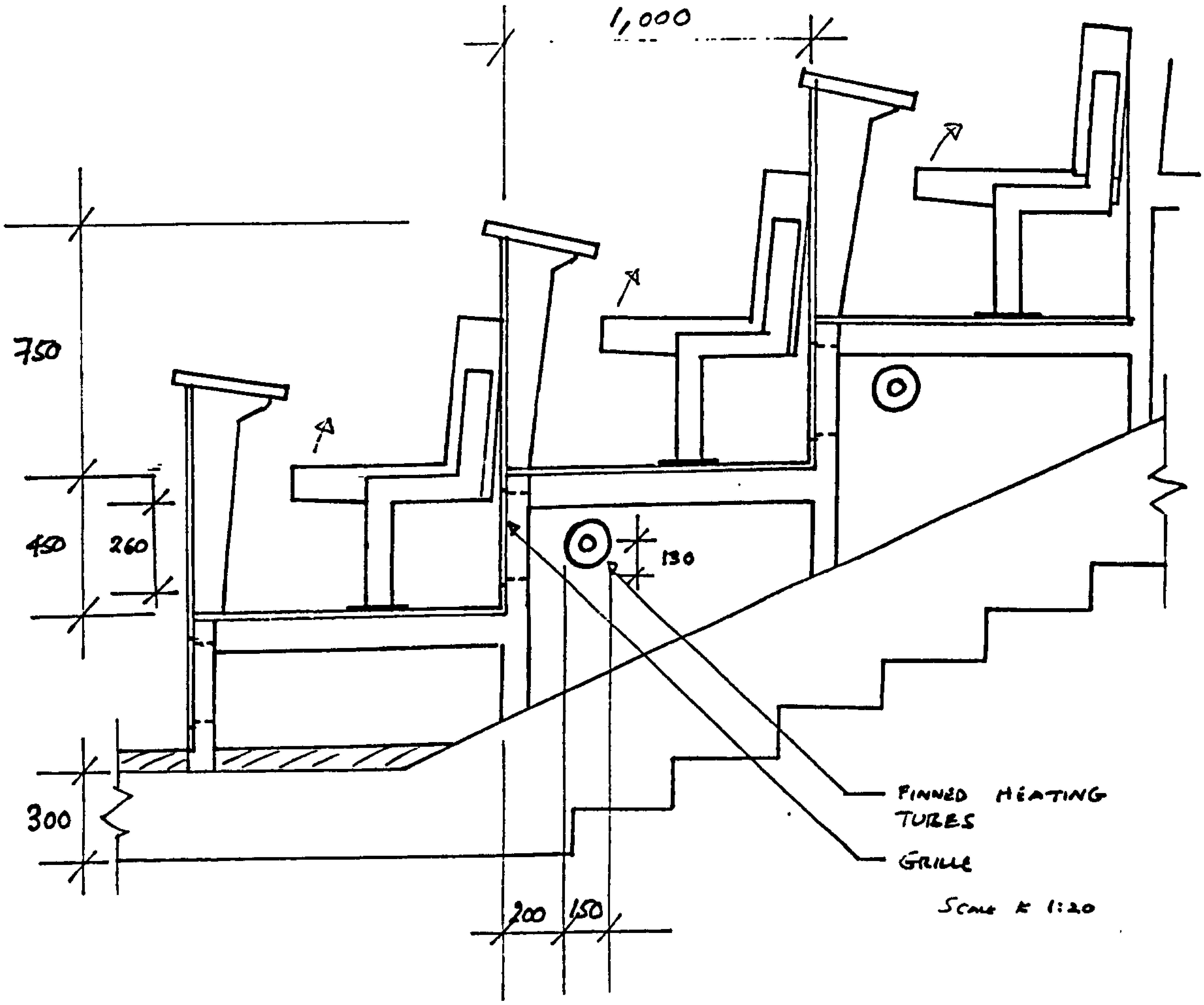
# Appendix B:

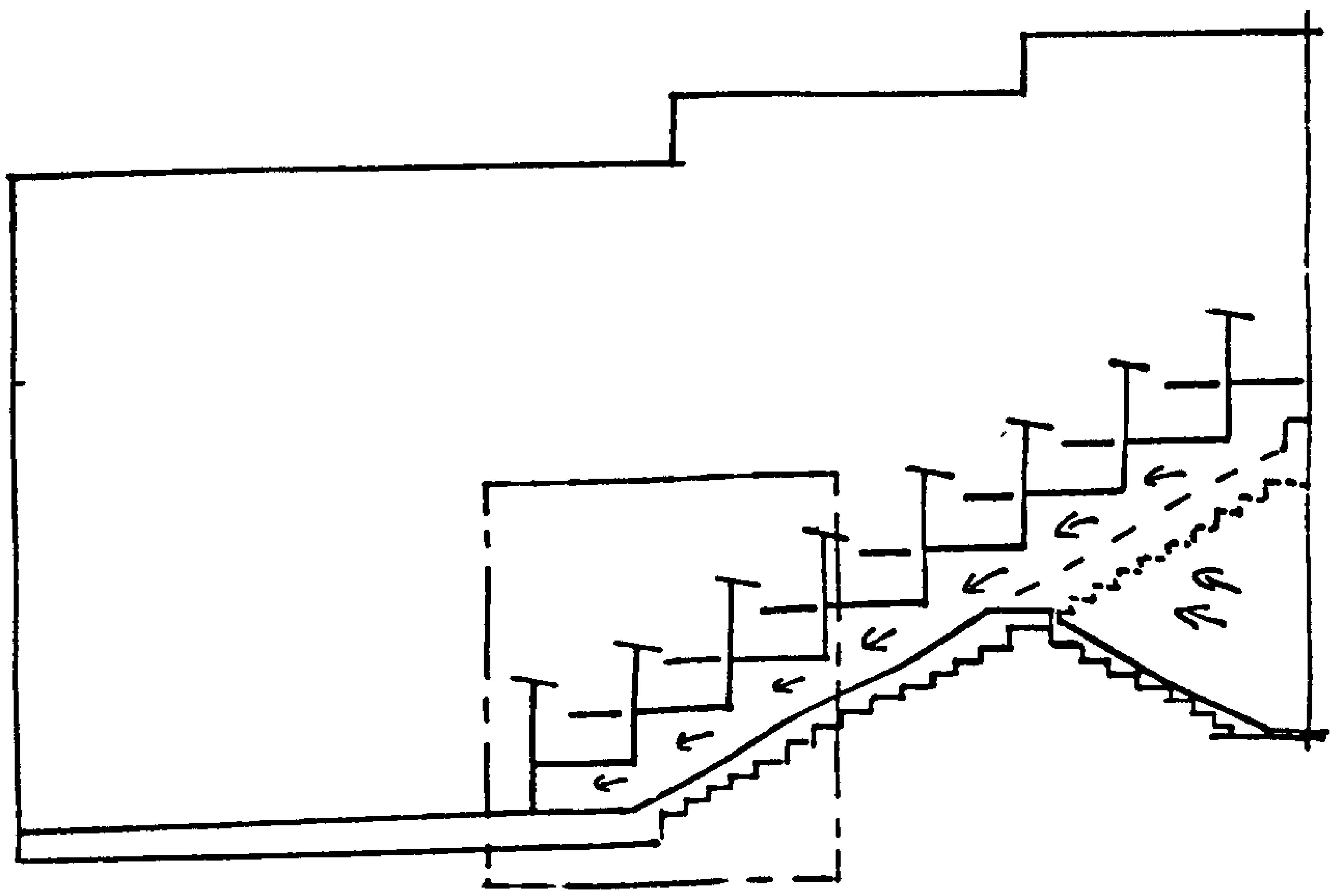
## Appendix B1 Data supplied to IEA Annex 26 modellers

## B1.1 Section and plan through theatre 1.10

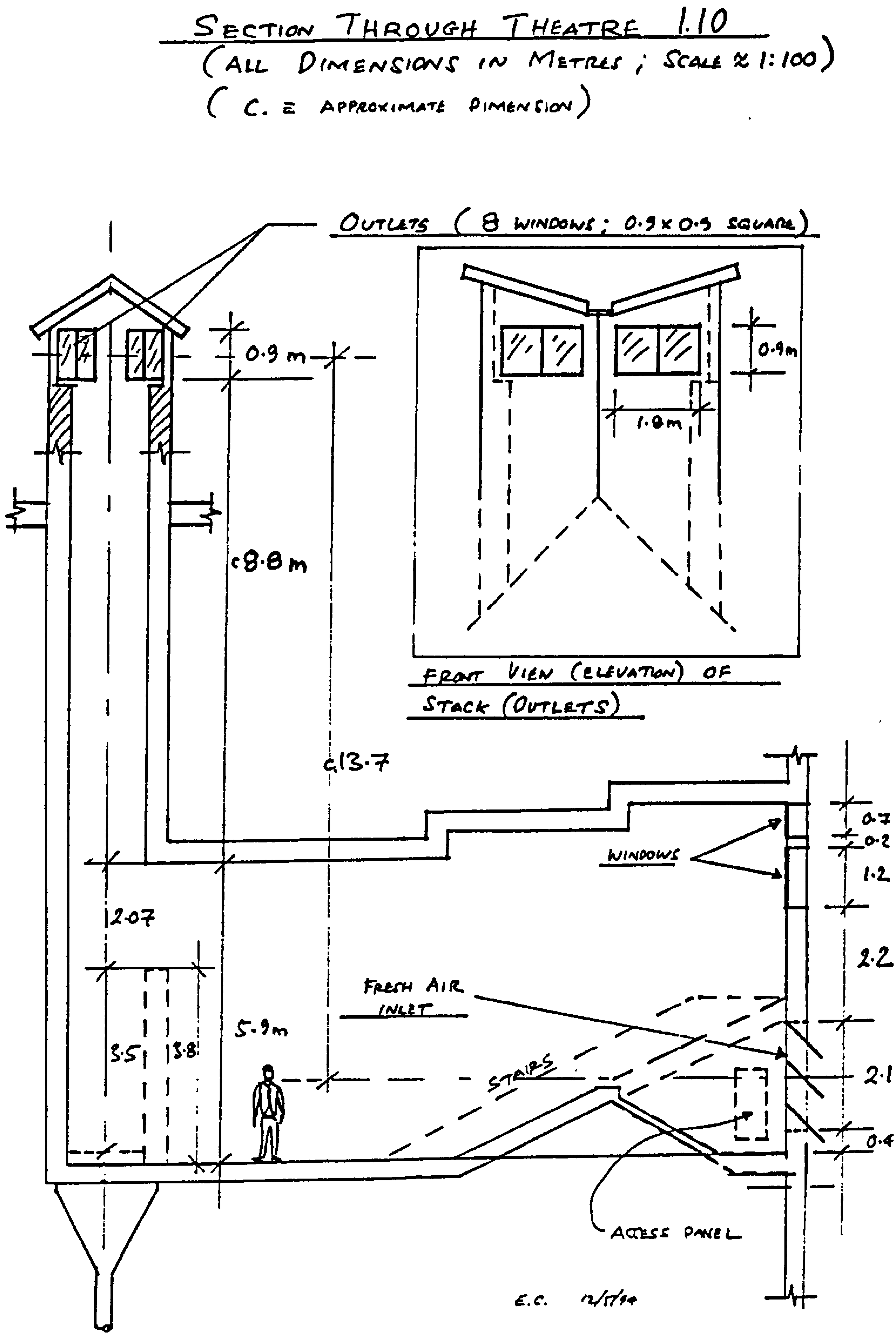


## B1.2 Section through theatre seating

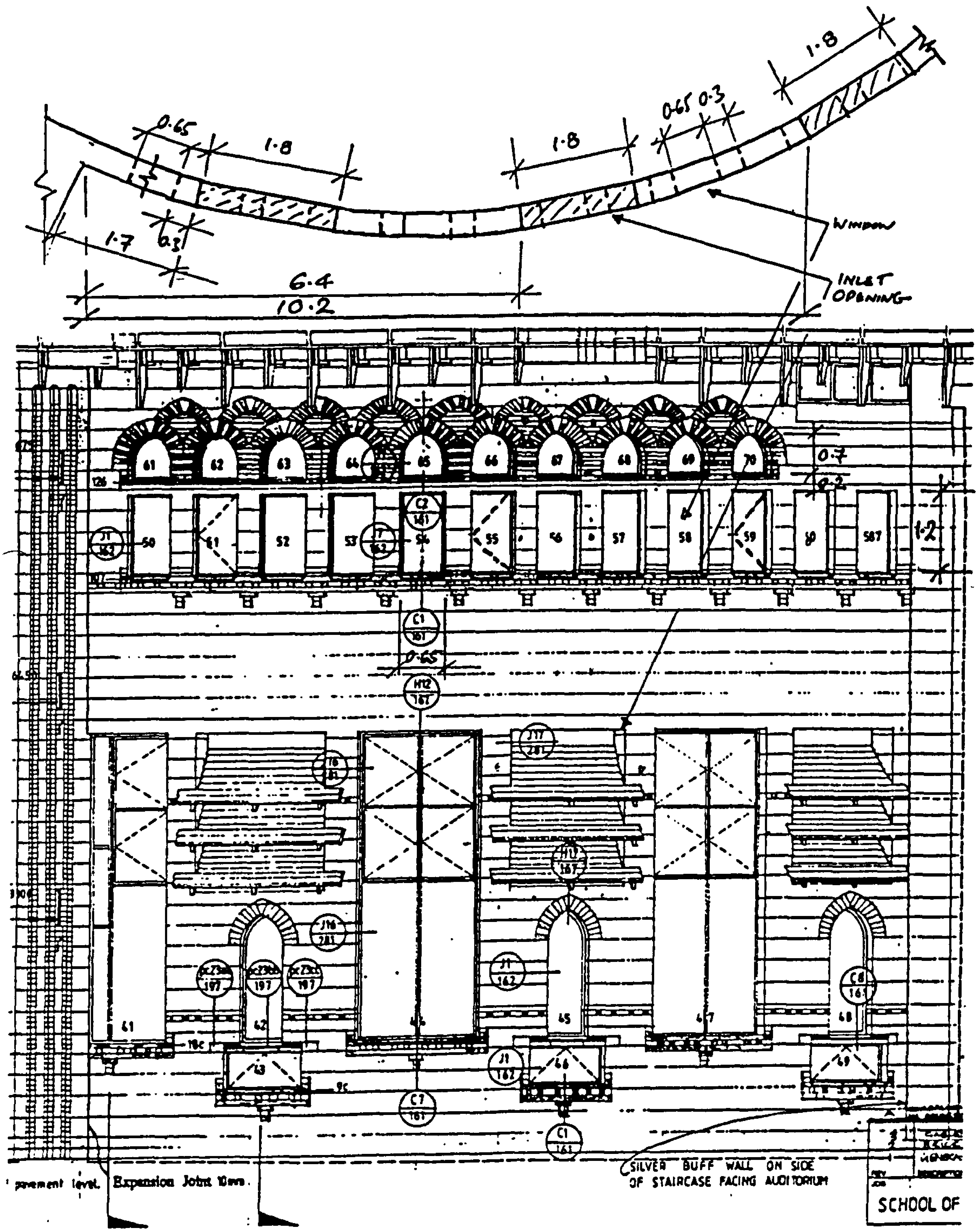


## B1.3 Section through theatre 1.10



## B1.4 Elevation of external facade, auditorium 1.10



## B1.5 Site data

| **Building** | **Queens Building**<br>**De Montfort University, Leicester** | |
|---|---|---|
| Room | Auditorium 1.10 | |
| | | |
| **Standard data** | | |
| | | |
| Site exposure (1) | Normal | |
| Surface emissivity | High | |
| Surface absorbtivity | Dark | |
| Latitude (2) | 51.7 | |
| Climate | Temperate | |
| Height above sea level (m) | 100 | |
| | | |
| Month | July | January |
| Sky cloudiness | 0.18 | 0.35 |
| Direct radiation factor | 0.72 | 0.88 |
| Diffuse radiation factor | 0.89 | 1.29 |
| Max. air temperature | 25 | 15 |
| Min. air temperature | 13 | -2.5 |

Table B1.5 Site data
Data used by consultants when designing the building

Note:

(1) Wind conditions for most suburban and country premises; 4th to 8th floors of city centre buildings.

(2) This is the latitude for London; it is assumed that the weather at Leicester is very similar to that recorded for Kew Gardens (West London).

## B1.6 Heat gains (auditorium 1.10)

| Heat gains (Auditorium 1.10) | | | | |
|---|---|---|---|---|
| Number | Heat gain type | Magnitude ($W\ m^{-2}$ or kW) | Location (1) | Profile (start/finish hours) |
| 1 | Occupants | Total (max.) = 15 kW (150 $W\ m^{-2}$ based on seating area of 100 $m^2$)<br><br>Average load = c. 5 kW (= 100 $W\ m^{-2}$ over 50 $m^2$) | 1.1 m above floor over seating area<br><br>a) uniformly distributed over one half of floor nearest to front of theatre<br>b) other layouts | 9:00 to 13:00<br><br>14:00 to 17:00 |
| 2 | Lighting | Total (max.) = 2.8 kW<br>Average = 15 $W\ m^{-2}$ based on floor area of 187 $m^2$ | Zone c. 0.2 m deep c. 1m below ceiling (actual zones used depends on occupancy layout) | As above |
| 3 | Equipment | Total (max.) = 500 W (i.e. one projector plus one over head projector) | Heat source area = c. 0.3 x 0.3 $m^2$ at positions at front and rear of theatre | As above |

Table B1.6 Heat gains

## B1.7 Data for the calculation of "U" values

| Building | Queens Building, Leicester | | | | | | | |
|---|---|---|---|---|---|---|---|---|
| Room | Auditorium 1.10 | | | | | | | |
| Construction No. | 1 | 2 | 3 | 4 | 5 | 6 | 7 | 8 |
| Name | Floor | Window | Wall | Door | Int wall | Ceiling | Stack wall | Roof |
| Ext. surface resistance ($m^2K\ W^{-1}$) | 0.1 | | | | | | | |
| *Layer 1* | | | | | | | | |
| Name | Concrete | Glass | Brick | Timber | Brick | Concrete | Brick | |
| Thermal onductivity ($W\ m^{-1}K^{-1}$) | 1.4 | | 0.84 | 0.14 | 0.84 | 1.4 | | |
| Thickness (m) | 0.15 | | 0.1 | 0.045 | 0.1 | 0.15 | | |
| Density ($kg\ m^{-3}$) | 2100 | | 1700 | 650 | 1700 | 2100 | | |
| Heat capacity ($J\ kg^{-1}K^{-1}$) | 840 | | 800 | 1200 | 800 | 1000 | | |
| *Layer 2* | | | | | | | | |
| Name | Cavity | Cavity | Cavity | | | Plaster | Mineral wool | |
| | | | | | | 0.16 | 0.04 | |
| Thermal resistance ($m^2K\ W^{-1}$) | 0.18 | | 0.1 | | | | | |
| Thickness (m) | 0.5 | | 0.15 | | | 0.01 | 0.05 | |
| Density ($kg\ m^{-3}$) | | | | | | 600 | 30 | |
| Heat capacity ($J\ kg^{-1}K^{-1}$) | | | | | | 1000 | 1000 | |
| *Layer 3* | | | | | | | | |
| Name | Timber | Glass | Glass fibre | | | | Chip board | |
| Thermal conductivity ($W\ m^{-1}K^{-1}$) | 0.14 | | 0.035 | | | | 0.15 | |
| Thickness (m) | 0.02 | | 0.1 | | | | 0.02 | |
| Density ($kg\ m^{-3}$) | 650 | | 25 | | | | 800 | |
| Heat capacity ($J\ kg^{-1}K^{-1}$) | 1200 | | 1000 | | | | 1200 | |
| *Layer 4* | | | | | | | | |
| Name | Carpet | | Brick | | | | | |
| Thermal conductivity ($W\ m^{-1}K^{-1}$) | 0.045 | | 0.84 | | | | | |
| Thickness (m) | 0.01 | | 0.1 | | | | | |
| Density ($kg\ m^{-3}$) | 160 | | 1700 | | | | | |
| Heat capacity ($J\ kg^{-1}K^{-1}$) | 1200 | | 800 | | | | | |
| Int. surface resistance ($m^2K\ W^{-1}$) | 0.1 | | 0.12 | 0.12 | 0.12 | 0.1 | 0.12 | |
| U value ($W\ m^{-2}K^{-1}$) | 1.16 | 3.4 | 0.29 | 1.99 | 2.78 | 2.71 | 0.60 | 0.31 |
| Y value ($W\ m^{-2}K^{-1}$) | 1.67 | 3.4 | 5.16 | 2.07 | 4.06 | | 1.28 | |
| Admittance time lag (h) | 1.95 | | 2.18 | 0.7 | 3.69 | | 4.09 | |
| Decrement factor | | | 0.38 | 0.99 | | | | |
| Decrement time lag (h) | | | 3.69 | 0.69 | | | | |
| Surface factor | 0.85 | 0.67 | 0.58 | 0.75 | 0.82 | | 0.94 | |
| Surface time lag (h) | 0.36 | | 2.33 | 0.23 | 1.93 | | 0.45 | |
| Mean solar gain factor | | 0.64 | | | | | | |
| Lightweight alt. solar gain factor (windows) | | 0.48 | | | | | | |

Table B1.7 Data for the calculation of "U" values

# Appendix B2

## Data for modelling the De Montfort Auditorium

1.0 Opening areas:

| BMS Setting (%) | Area ($m^2$) of Inlets | Inlet grilles | Stack openings | Outlets (stack top) |
|---|---|---|---|---|
| 50 | 2.42 | 14.69 | 6.11 | 4.64 |
| 75 | 3.19 | 14.69 | 6.11 | 5.62 |
| 100 | 3.57 | 14.69 | 6.11 | 7.49 |

Table B2.1(a) opening areas

| Opening | Width (m) | Height (m) | Free Area Fraction | Area ($m^2$) | Number | Total Area ($m^2$) |
|---|---|---|---|---|---|---|
| Inlets | 1.8 | 1.97 | 0.34 | 1.21 | 3 | 3.57 |
| Inlet grilles | 90.4 | 0.25 | 0.65 | 14.69 | | 14.69 |
| Stack openings | 1.56 | 1.96 | 1.0 | 3.055 | 2 | 6.11 |
| Outlets | 0.9 | 0.52 | 1.0 | 0.468 | 8x2 | 7.49 |

Table B2.1(b)

Note: BMS = Building Management System
Free area percentages :
Inlets; 80 %; Grille inlets; 65 %
Stack opening; 100 %, Outlets (stacks); 100 %
(refer to section and plan view of theatre supplied in May 1994)

## 2.0 Measurement data

### 2.1 Winter runs

Date: 21/12/94

| Time | | Opening | ACR | Tavg (1) | Tceil1 | Tceil | Twall1 | Twall | Text |
|---|---|---|---|---|---|---|---|---|---|
| Start | Finish | (%) | (1/hr) | (int, °C) | (°C) | (avg, °C) | (°C) | (ext, °C) | (°C) |
| 16:51 | 17:04 | 50 | 22.3 | 21.2 | 20.7 | 20.6 | 21.2 | 18.6 | 3.3 |
| 17:38 | 17:48 | 75 | 28.2 | 20.5 | 20.6 | 20.5 | 21.1 | 18.4 | 3.2 |
| 18:24 | 18:32 | 100 | 29.5 | 19.8 | 20.1 | 20.0 | 20.4 | 17.8 | 4.4 |

Table B2.2 Winter runs

Notes:
Heating: off
Occupant heat gains = 5 kW

Date: 22/12/94

| Time | | Opening | ACR | Tavg (1) | Tceil1 | Tceil (avg, °C) | Twall1 | Twall | Text |
|---|---|---|---|---|---|---|---|---|---|
| Start | Finish | (%) | (1/hr) | (int, °C) | (°C) | | (°C) | (ext, °C) | (°C) |
| 10:00 | 10:23 | 50 | 22.0 | 19.5 | 19.6 | 19.6 | 19.9 | 17.2 | 2.2 |
| 17:38 | 17:48 | 75 | 29.0 | 19.7 | 19.1 | 19.1 | 19.4 | 16.6 | 4.3 |
| 18:24 | 18:32 | 100 | 28.2 | 19.5 | 19.3 | 19.3 | 19.6 | 16.8 | 4.7 |

Table B2.3 Further winter runs

Notes: wind speeds low (< 6 m/s) at 18.5 m
Heating: on
Occupant heat gains = 5 kW

### 2.2 Mid-season runs

Date: 4/2/95

| Time | | Opening | ACR | Tavg | Tceil1 | Tceil (avg, °C) | Twall1 | Twall | Text |
|---|---|---|---|---|---|---|---|---|---|
| Start | Finish | (%) | (1/hr) | (int, °C) | (°C) | | ( °C) | (ext, °C) | (°C) |
| 12:46 | 12:49 | 50 | 18.8 | 19.0 | 20.0 | 19.9 | 20.4 | 17.7 | 11.3 |
| 13:15 | 13:21 | 75 | 23.0 | 18.8 | 20.0 | 19.9 | 20.4 | 17.7 | 10.9 |
| 13:40 | 13:52 | 100 | 24.2 | 18.0 | 19.9 | 19.8 | 20.3 | 17.6 | 11.1 |

Table B2.4 Mid-season runs

Notes: wind speeds low (< 6 m/s) at 18.5 m
Heating: off
Occupant heat gains = 5 kW

Notation:

| | |
|---|---|
| Occ. | Occupants |
| e | Estimate |
| ACR | Air change rate (average) (1/hour) |
| Tavg(int) | Average room temperature (°C) |
| Text | Average external temperature (°C) |
| Tceil1 | Ceiling surface temperature near internal wall (°C) |
| Tceil (avg) | Calculated average ceiling slab temperature (°C) |
| Twall1 | Internal wall surface temperatures (brick) (°C) |
| Twall (ext) | Average inner surface temperature of external wall (°C) |

## 2.3 Occupancy heat gain distributions

### 2.3.1 Summer heat load layout:

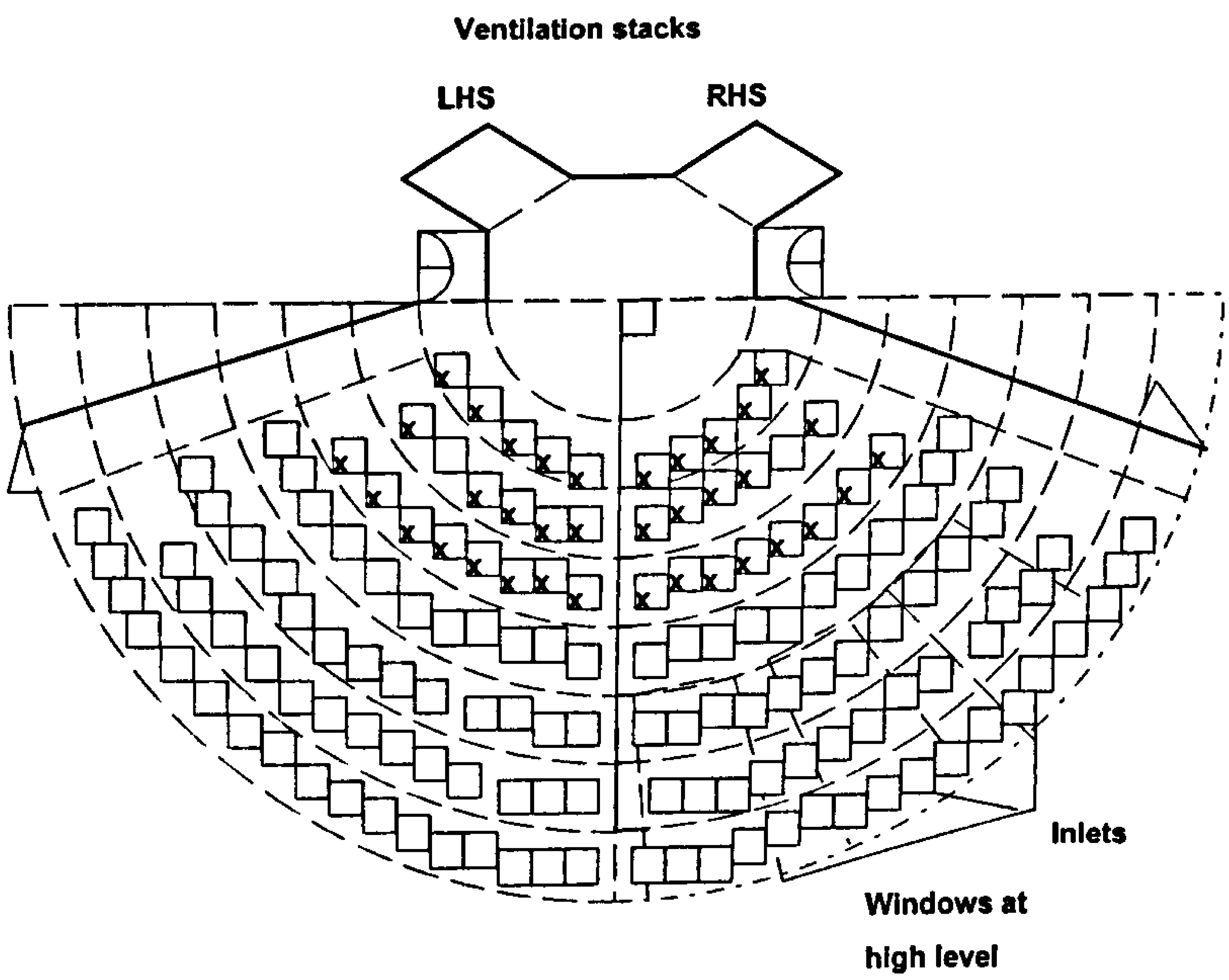



Figure B2.1 Summer heat load layout:

2.3.2 Winter/mid-season heat load layout:

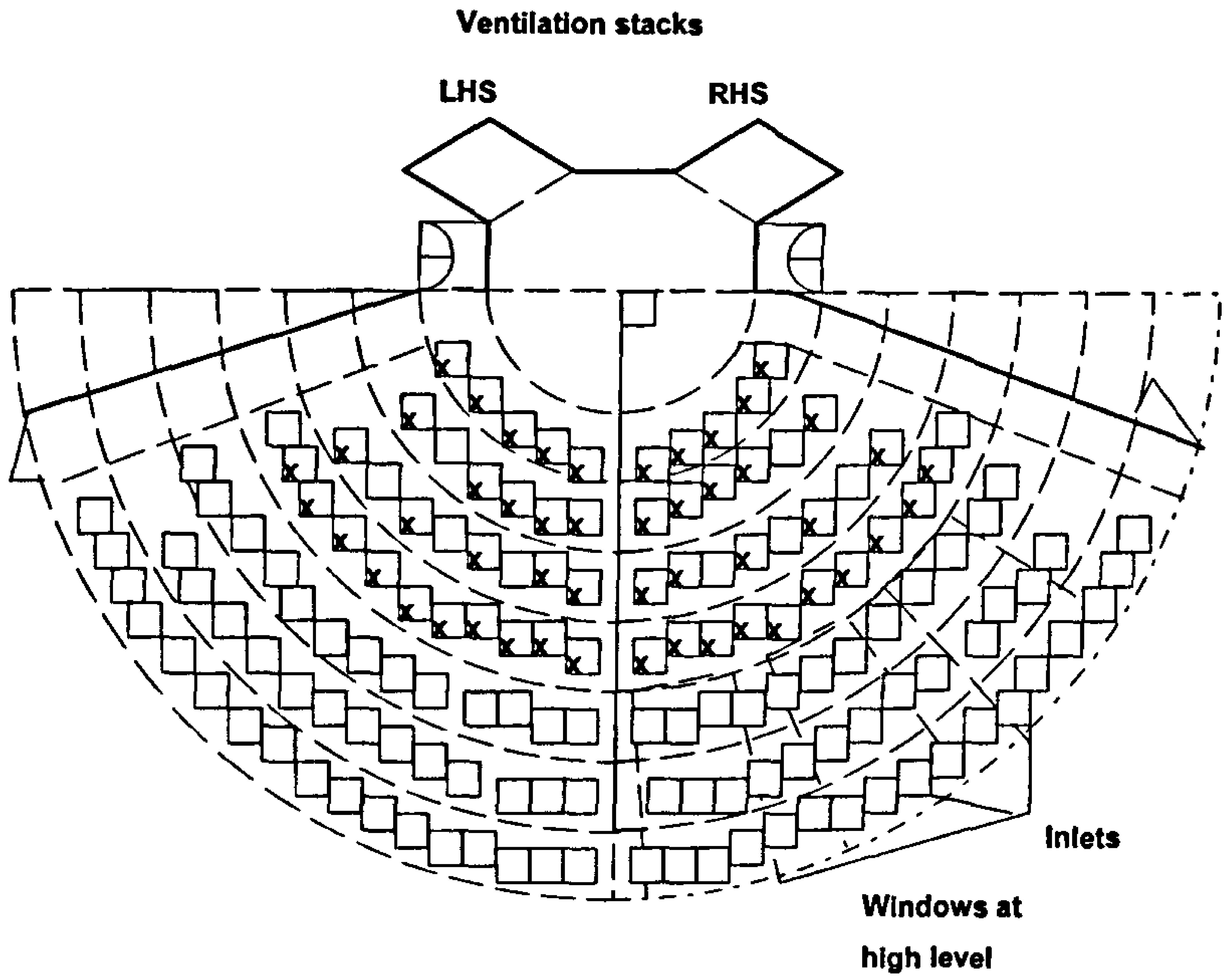



Figure B2.2 Winter/mid-season heat load layout:

2.4 Location of points of interest:
2.4.1 At height of 9.25 m within centre of stacks above auditorium floor
2.4.2 0.5 m from inlet grille, 0.17 m above floor, fifth row (underside of seat on right hand side; right hand side of theatre) (i.e. a position directly in line with inlet plenums).
2.4.3 Height(s) of stratfication zone (s) (?) (below ceiling slab)

2.5 Predicted results required
2.5.1 Steady state simulations:

2.5.1.1 Overall air change rates for stated conditions

2.5.1.2 Local air change rates in
- i) Stacks
- ii) LHS (i.e. the LHS volume of the theatre)
- iii) RHS (as for ii)
- iv) LHS (high level)
- v) RHS (high level)

2.5.1.3 Air speeds at points given in 2.4 above
LHS; RHS Left and right hand sides (facing the front of the theatre)

2.5.2 Transient simulations

2.5.2.1 Prediction of downflows in stacks when outlets start to close (i.e. start with results of steady state simulation then vary inlet and outlet areas with time; e.g. at t = 0 s, A/Amax = 100 %, at t = 60 s; A/Amax = 50 %, at t = 120 s; A/Amax = 0 %)
s = seconds
A/Amax = ratio of actual opening area to maximum opening area

2.5.2.2 Prediction of variable air change rates for varying wind speeds and directions (e.g. Carry out runs with wind speeds increasing from 3 m/s to 6 m/s in a period of (say) 5 s; change wind direction from East to West using a wind speed of 3 m/s)

2.5.2.3 Determination of critical wind speed(s) at which wind induced pressure difference equals that produced by the stack effect.

Note: 2.5.2.2 and 2.5.2.3 may be too difficult to achieve at this time and further experimental runs are required.

# Appendix C

## Measurement data files for auditorium 1.10; Queens Building, Leicester.

### List of directories and description of contents:

54N10 Dantec basic files. Used for processing data generated by the Dantec logging system (air speeds).

File format: (summer 1994) a_ddmmv#.vel
(winter 1994 to summer 1995)) a1ddmmv#.vel
a1ddmmv#.tmp
(*.vel; air speeds; *.tmp; air temperatures).

AUTOV avt_6s.bas; program for processing autovent data files (i.e. type: "Qbasic", then name of BASIC program file. This program inputs an Autovent "*.dta" file and outputs spreadsheet compatible files (*.csv)) bms_frm.bas; program for changing the format of Building Management System files (pxy_mm_yy.txt; qxy_mm_yy.txt); outputs spreadsheet compatible file.

BMS Building management system data files. These are contained in directories that are named after the date on which data was downloaded (the downloading program (eoin.exe) can be run at the BMS terminal every six days; this copies data obtained from BMS sensors onto a floppy disk. Allow about 30 minutes for completion of this process. (Program file, bms_frm.bas used for generating spread-sheet compatible files giving data for 14 parameters (internal and external temperatures, opening positions etc.)) (see note below)

CSV: Comma separated files (*.csv) files (tracer gas concentrations; R110_xyz.csv, surface temperature files; sddmmyy.*, external temperature files; addmmyy.*, air velocity files; addmmyv#.*), and Supercalc 5 spreadsheet files (*.cal)

DTA Raw tracer gas concentration data files (*.dta), generated using Autovent

SCW Supercalc for windows files
File type: *.mds.

TREND: Files for configuring and operating the trend process logger (used for logging signals (0 to 5 V) produced by the Furness pressure sensors (0-10 Pa). The logger can monitor up to 7 input signals.

WDATA Weather data; i.e. wind speed and direction (spreadsheet compatible files) obtained using Squirrel logger.

WORD: Document (Word 6.0) files, annex 26 reports, letters, papers

| Parameters measured | File names | Data file type | Dates applicable |
|---|---|---|---|
| Air speeds | a1ddmmv#.vel<br>a1ddmmv#.tmp<br>addmmyv#.vel<br>addmmyv#.tmp | Air speeds<br>Air temperatures<br>Air speeds<br>Air temperatures<br>(files spread-sheet compatible | 7/94 to 6/95<br><br>6/95 to 2/96 |
| Tracer gas concentrations | r110_xyz.dat<br>r110_xyz.csv | Raw data; text file<br>Gas concentrations; spread-sheet compatible | 8/94 to 7/96 |
| BMS point values | pxy_m_yy.txt<br>qxy_m_yy.txt<br>bms_ddmm.csv<br><br>b_ddmmyy.csv | Data for "point" no. xy<br><br>Data for up to 14 "points"; spread-sheet compatible<br>Data for up to 14 "points"; spread-sheet compatible | 8/94 to 12/95<br><br>to 12/94<br><br>1/95 to 12/95 |
| Air and surface temperatures (Squirrel logger data) | sddmmyy.cal<br><br><br>addmmyy.cal | Surface temperatures of walls and ceiling slab (4 pts)<br>External air temperatures | 7/94 to 7/96<br><br><br>7/94 to 7/96 |
| Air velocities (ultra-sonic velocity data) | ddmmyyl1.dat<br><br>ddmmyyr1.dat | "x", "y", "z" velocities at entrance to lhs stack<br>"x", "y", "z" velocities at entrance to rhs stack | 8/95 |
| Wind speed and direction | wddmmyy.csv | Wind speed and direction at 5 minute intervals (at location on roof of Gateway house, h = c. 26m) | 9/95 to 7/96 |

Table C1 Formats of data files used
Note: data is available only for specific dates and times.
ddmmyy = date; month; year, e.g. 090296

Note on BMS data files:

The following files are currently stored (sampling interval = 15 minutes):

| **File name:** | **Description:** |
|---|---|
| P00_7_95.txt | Outside air temp |
| P01_7_95.txt | Outside humidity |
| P04_7_95.txt | Relative wind direction (0-10); data not reliable |
| P05_7_95.txt | Average wind speed (mph) (data not reliable) |
| P06_7_95.txt | Rain sensor (0 = no rain) |
| P08_7_95.txt | Stack air temperature (RHS, Top) |
| P09_7_95.txt | Stack air temperature (RHS, intermediate pos'n) |
| P10_7_95.txt | Stack air temperature (RHS, bottom) |
| P11_7_95.txt | Inlet plenum air temp |
| P12_7_95.txt | Room temperature (sensor #2) |
| P13_7_95.txt | Room temperature (sensor #3) |
| P14_7_95.txt | Room temperature (sensor #4) |
| P15_7_95.txt | Heating valve position |
| P16_7_95.txt | Air velocity (RHS stack), not reliable |
| P17_7_95.txt | CO2 concentration (RHS stack), not reliable |
| P18_7_95.txt | Fan on/off |
| P19_7_95.txt | Room condition; plant enabled/disabled (on/off) |
| P20_7_95.txt | Fresh air damper positions (%) |
| P21_7_95.txt | Outlet positions (top of stacks) |
| P22_7_95.txt | Average room temperature |
| P23_7_95.txt | Stack air temperature (LHS, intermediate pos'n) |
| P24_7_95.txt | 1.12; Stack air temperature (RHS, top) |
| P25_7_95.txt | 1.12; Stack air temperature (RHS, intermed. pos'n) |
| P26_7_95.txt | 1.12; Stack air temperature (RHS, bottom) |
| P27_7_95.txt | 1.12; Stack air temperature (LHS, top) |
| P28_7_95.txt | 1.12, air velocity, RHS stack (not reliable) |
| P29_7_95.txt | 1.12 CO2 level, RHS stack (not reliable) |
| P30_7_95.txt | 1.12, fan on/off |
| P31_7_95.txt | 1.12, Room condition; plant enabled/disabled (on/off) |
| P32_7_95.txt | 1.12, fresh air damper positions (%) |
| P33_7_95.txt | 1.12, outlet positions (%) |
| P34_7_95.txt | 1.12, average room temperature |
| P35_7_95.txt | 1.12; Stack air temperature (LHS, intermed. pos'n) |
| P36_7_95.txt, P39_7_95.txt | 1.12; heating valve position |
| P37_7_95.txt | 1.12, underfloor plenum air temperature |
| P38_7_95.txt | 1.12, heating circulation pump (?) |
| P61_7_95.txt | 1.12, air temperature, sensor #3 |
| P62_7_95.txt | 1.12, air temperature, sensor #4 |
| P63_7_95.txt | 1.12, air temperature, sensor #5 |
| Q00_7_95.txt | Ceiling slab temperature |
| Q01_7_95.txt | Air temperature, sensor #1 |
| Q02_7_95.txt | 1.12; Ceiling slab temperature |
| Q03_7_95.txt | 1.12; Air temperature, sensor #1 |

| Run | Date | Time | | Opening Settings (%) | Heat Load (OC) (kW) | File names |
|---|---|---|---|---|---|---|
| | | Start | Finish | | | |
| | | | | | | |
| W1 | 21-Dec-94 | 16:51 | 17:04 | 50 | 5 | a12112v2.vel,tmp |
| | | 17:38 | 17:48 | 75 | 5 | a12112v3.vel,tmp |
| | | 18:24 | 18:32 | 100 | 5 | r110_048.* |
| W2 | 22-Dec-94 | 10:00 | 10:23 | 50 | 5 | a12212v1.vel,tmp |
| | | 10:49 | 11:05 | 75 | 5 | r110_049.* |
| | | 11:40 | 12:05 | 100 | 5 | bms_2312.csv |
| W3 | 04-Jan-95 | 11:12 | 11:25 | 50 | 5 | a10401v1.vel,tmp |
| | | 11:49 | 12:01 | 75 | 5 | r110_050.* |
| | | 12:39 | 13:37 | Infiltration | 5 | b_060195.cal |
| W4 | 12-Jan-95 | 14:05 | 14:24 | Infiltration | var | r110_052.* |
| | | 14:24 | 15:00 | | var | b_180195.* |
| M1 | 09-Jan-95 | 09:31 | 17:49 | Infiltration | var | r110_051.* |
| | | 15:01 | 16:01 | | var | |
| | | 16:01 | 17:01 | | var | |
| M2 | 19-Jan-95 | 11:59 | 12:17 | 50 | var | a119015t.xls |
| | | 14:59 | 15:59 | Infiltration | var | a1190195.vel,tmp |
| | | | | | | r110_053.* |
| | | | | | | b_240195.cal |
| M3 | 28-Jan-95 | 13:18 | 13:21 | 50 | 5 | a1280195.vel,tmp |
| | | 14:06 | 14:15 | 75 | 5 | r110_054.* |
| | | 14:51 | 15:00 | 100 | 5 | b_300195.csv |
| M4 | 03-Feb-95 | 16:42 | 16:52 | 50 | 5 | a1030295.vel |
| | | 17:13 | 17:21 | 75 | 5 | |
| | | 17:51 | 17:58 | 100 | 5 | r110_055.* |
| M5 | 04-Feb-95 | 12:46 | 12:49 | 50 | 5 | a1040295.vel |
| | | 13:15 | 13:21 | 75 | 5 | r110-056.* |
| | | 13:40 | 13:52 | 100 | 5 | b_070295.csv |
| | | | | | | s1102951.csv |
| M6 | 09-Mar-95 | 18:01 | 18:15 | 0 | Var | a_090395.cal |
| | | | | | | a100395.cal |
| | | | | | | r110_057.csv |
| M7 | 14-Mar-95 | 10:00 | 11:00 | 0 | Var | a140395.cal |
| | | | | | | rr10_059.csv |
| M8 | 15-Mar-95 | 13:00 | 13:00 | 0 | Var | b150395.* |
| | | | | | | b160395.* |
| | | | | | | a150395.* |
| | | | | | | r110_060.csv |
| M9 | 23-Mar-95 | 14:27 | 14:35 | 0 | Var | b_280395.* |
| | | | | | | r110_061.csv |

Table C2 Experimental data file names

| Run | Date | Time Start | Time Finish | Opening Settings (%) | Heat Load (OC) (kW) | File names |
|---|---|---|---|---|---|---|
| | | | | | | |
| M10 | 11-Apr-95 | 10:40 | 10:44 | 50 | 5 | a110495.cal |
| | | 11:12 | 11:18 | 75 | 5 | B_130495.* |
| | | 11:34 | 11:42 | 100 | 5 | r110_062.csv |
| M11 | 12-Apr-95 | 10:56 | 11:04 | 50 | 5 | a120495.cal |
| | | 11:28 | 11:38 | 75 | 5 | r110_063.csv |
| | | 11:56 | 12:00 | 100 | 5 | |
| S1 | 11-Apr-95 | 13:38 | 13:44 | 50 | 5 | r110_062.csv |
| | | 14:00 | 14:08 | 75 | 5 | |
| | | 14:24 | 14:32 | 100 | 5 | |
| S2 | 02-May-95 | 16:42 | 17:00 | Infilt | 0 | a020595.cal |
| | | 17:14 | 17:24 | 50 | 0 | b_020595.* |
| | | | | | | r110_064.csv |
| S3 | 04-May-95 | 14:36 | 14:48 | 100 | 0 | a040595.cal |
| | | | | | | b_090595.csv |
| | | | | | | s12may95.* |
| | | | | | | r110_065.csv |
| S4 | 02-Aug-95 | 10:42 | 11:04 | 100 | 5 | a020895.cal |
| | | 11:12 | 11:38 | 100 | 5 | s020895.csv |
| | | 14:32 | 15:00 | 100 | 5 | r110_082.* |
| S5 | 03-Aug-95 | 10:49 | 11:15 | 100 | | a030895v5.vel,tmp |
| | | 11:25 | 11:53 | 100 | | |
| | | 14:07 | 14:33 | 100 | | |
| | | 15:23 | 15:51 | 100 | | r110_083.* |
| S6 | 08-Aug-95 | 11:29 | 11:43 | 50 | | a08085v3.cal |
| | | 12:33 | 12:47 | 100 | | s080895.cal |
| | | 15:25 | 15:49 | 100 | | r110_084.* |
| S7 | 09-Aug-95 | 09:37 | 11:45 | Infiltration | | a09085v2,v3.vel,tmp |
| | | 15:19 | 15:33 | 50 | | r110_085.* |
| | | | | | | b_100895.cal |
| | | | | | | s1408001.* |
| S8 | 17-Aug-95 | 12:09 | 12:37 | Infiltration | | a17085v1.vel,tmp |
| | | 16:01 | 17:21 | Infiltration | | s180895.cal |
| | | | | | | S2308001.* |
| | | | | | | r110_090.* |

Table C2 Experimental data file names (continued)

# Appendix D

## Appendix D1 Measurement of Air Change Rates using a Tracer Gas - Methodology

1) Experimental procedure

a) Install, check, and calibrate equipment:

- Autovent plus software
  (check flowrates to/from sampler with bubble flow meter)
- Tubing (check for leakages, kinks etc.)
- Data loggers (initial settings)
- Analyser: i) zeroing using $N_2$
  ii) check full scale deflection using span gas
- Sensors (initially group sensors together to check that the same readings are obtained)
  - Temperature sensors
  - Pressure sensors
  - Velocity sensors

Check that weather data is being recorded

b) Check and record positions of inlet dampers, outlets, and windows. Ensure doors and windows are initially closed.

c) Try to maintain uniform conditions in areas surrounding the theatre in question (e.g. maintain set point temperatures in these areas).

d) Check sensor positions. Check the availability of power supplies to equipment, including fans.

e) Record experiment start time.

f) Switch on fans, loggers, computer(s) and other equipment.

g) Start Autovent control software.

h) Connect gas supply line(s) to the Autovent injection units.

i) Wait for complete mixing of tracer gas in the room (c. 10 minutes).

j) Switch off fans (if possible do this without entering the space).

k) Set inlet and outlet openings (including windows) such that the following areas are obtained:

   i) inlet opening area = $m^2$

   ii) outlet opening area = $m^2$

l) Commence logging of data.

m) Terminate experiment after c. 60 minutes (this is a short enough time for external conditions to remain essentially constant, i.e. constant air temperatures, wind velocities and directions, and long enough to collect sufficient data to generate a satisfactory plot of the natural log of tracer gas concentration versus time (hours).

n) Record finish time.

o) Download data from datalogger(s) for the above period; retrieve weather data (air temperatures, wind velocities and directions, or corresponding voltage readings) from B.M.S. (building management system). Convert data to the appropriate units if required.

p) Shut down equipment.

q) Load data into separate spreadsheet files

- Tracer gas concentrations and corresponding times
- Room air temperatures, sensor locations and times recorded
- Room air velocities, sensor locations and times recorded
- Stack pressures (convert to flowrates); times
- Slab surface temperatures (both sides); times
- Weather data i) air temperatures
  ii) wind velocities
  iii) wind directions
  iv) corresponding times

r) Repeat for other inlet and outlet areas.

## Appendix D2 List of equipment used

1. Autovent injection and sampling system including PC plus software.
2. Autovent thermistor temperature sensors (12 off) plus associated cabling.
3. Tracer gas ($SF_6$) and $N_2$ cylinders.
4. Plastic 6 mm tubing.
5. Mixing fans.
6. Infra-red Gas Analyser for the measurement of $SF_6$ and $CO_2$ gas concentrations (full scale resolutions = 50 ppm).
7. Squirrel temperature loggers (2 off); external (2 off) and surface temperature (4 off) sensors.
8. Dantec low velocity anemometry system; Dantec recorder, PC plus software, cabling, 24 sensors.
9. Ultra-sonic system; sensors, PC's plus software (one per sensor), cabling.
10. Hand-held air speed probe.
11. Smoke generator.
12. Pressure transducers (2 off) plus power supply.
13. 100 W tungsten filament light bulbs plus bulb holders, mounting and associated cabling ( heat sources).
14. Access ladder (2 sections).

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# Appendix F Participation in IEA Annex 26.

## F1 Program objective

In 1991, the International Energy Agency (IEA) decided to initiate a new collaborative research program entitled "Energy Efficient Ventilation of Large Enclosures". The main objective of this program was to develop methods to minimise the energy consumption of large enclosures while ensuring that comfort conditions and good indoor air quality were being maintained in all occupied areas. These methods would comprise both experimental and computational techniques and would be developed by using the techniques on a number of case study buildings (see table F1.1).

A large enclosure may be defined in a number of ways, i.e.;
(1) the occupied zone is a small fraction of the total volume,
(2) the Rayleigh number, Ra (=GrPr), is large, and
(3) the thermal buoyancy has a dominant effect on air motion.

The range of volumes of enclosures examined was very large, the smallest being about 1,000 $m^3$ (the De Montfort Auditorium) to the largest with a volume of 114,000 $m^3$ (Gjovik Olympic Mountain Hall).

## F2 Outcome

The final outcome of this research program would be the publication of a key report. The major sections of this report would be i) "Design Guide" (editor; Bob Waters); ii) "Case Studies" (editors; Dirk Muller and Nobert Vogl); and iii) "Analysis and Prediction Techniques" (editor; Per Heiselberg). In addition to the IEA report, the technical work is published in numerous publications in scientific journals, as conference proceedings, and in technical reports of the contributing institutes.

## F3 Input provided by De Montfort University

The main input provided by De Montfort University (through the author and his colleagues) was to collect and process experimental data for the Queens Building auditorium, which acted as one of the key case study buildings being examined. In addition, a section of the "Analysis and Predictions Techniques" report was be written and detailed information was provided for the "Case Study" report (see above). Other work included providing experimental data for other researchers for comparison with zonal and CFD (computational fluid dynamics) simulations. Free exchange of data was performed using the internet. A directory had been set up by the operating agent for facilitating the above work. See appendix for information provided on the auditorium.

## F4 List of countries, case studies and other information

Participants in 12 countries were involved in the work of Annex 26. The countries were; Denmark, Finland, France, Germany, Italy, Japan, the Netherlands, Norway, Poland, Sweden, Switzerland and the United Kingdom.

A list of the case study buildings is shown in table F1.1.

| No. | Building | Description | Main reporter |
|---|---|---|---|
| 1 | De Montfort University Auditorium Leicester UK | Naturally ventilated auditorium; floor area = 190 $m^2$, Volume = 870 $m^3$. No. occupants = 150 | Eoin Clancy Coventry University CV1 5FB, UK e.clancy@cov.ac.uk |
| 2 | Experimental Atrium Japan | Atrium; 4.3 m x 7 m x 4.5m (Japan) | S. Kato Unversity of Tokyo kato@iis.u-tokyo.ac.jp |
| 3 | Gjovik Olympic Mountain Hall Gjovik Norway | Stadium within mountain; vol. = 114,000 $m^3$ | H. Mathisen SINTEF Trondheim, Norway hans.m.mathisen@energy.sintef.no |
| 4 | Fibre Glass Reinforced Plastics Factory Caen France | Factory; vol. = 35 m x 42 m x 6 m = 8,820 $m^3$ | S. Collineau INRS, France collineau@inrs.fr |
| 5 | Turnhalle Munchen Germany | Gymnasium vol. =15 m x 27 m x 5.5 m = 2,228 $m^3$ | N.Vogl LWK, Aachen, Germany vogl@wuek.RWTH-Aachen.DE |
| 6 | Grafenau Zug Atrium Zug Switzerland | Atrium between office buildings | A. Schaelin, ETH Zurich, Switzerland schaelin@iet.mavt.ethz.ch |
| 7 | Polytechnic Auditorium Italy | Lecture theatre | M Perino, Turin, Italy marco@polen2.polito.it |

Table F1.1. List of main case study buildings

## F5 Procedure adopted by IEA Annex 26 participants

The approach was, during the preparation stage, to identify the indoor climate problems encountered in actual large enclosures. A large number of existing case study buildings

were selected and energy efficient, ventilation strategies and problems were studied. Based on the survey of real case studies, methods for analysis and design were developed.

The study of the movement of air was carried out using field measurements, laboratory experiments and computer simulations. Simple computer models, or engineering procedures may be performed by the designer himself. Complex numerical techniques to calculate the details of the entire flow and temperature fields require specialised skills and adequate computer resources. However the need for "on-site" expertise and powerful computers is fast diminishing. Existing CFD codes will now happily run on personal computers (previously the reserve of expensive workstations). However, the very large number of cells required for the accurate modelling of large spaces means that workstations are usually required. Field models (e.g. CFD simulations) have been tested in smaller rooms via IEA Annex 20 ("Air Flow Patterns Within Buildings"; the author was a participant in the work of this annex). The performance and reliability of CFD codes have improved significantly over the past few years, and CFD techniques are now an important and accessible tool for designing the ventilation systems of large spaces.

The annex 26 program was adopted by the executive committee of the Energy Conservation in Buildings and Community Systems Implementation Agreement in November 1991. The 1 year preparation phase started in April 1, 1992. Over this initial 12 month period, the required tasks were defined, the project was structured, and a review of the current state of the art was carried out. No firm commitments were required for that phase. The working phase lasted from 1 April 1993 until July 31 1996. The Roomvent '96 conference marked the completion of the project.

During the course of the project, participants exchanged technical information, co-ordinated the work and planned the final report at semi-annual expert meetings. The meetings are listed in table F1.2.

| Number | Location | Date | Attended by DMU representative (yes/no) |
|---|---|---|---|
| 1 | Aalborg, Denmark | August 1992 | |
| 2 | Aachen, Germany | March 1993 | |
| 3 | Poitiers, France | October 1993 | |
| 4 | Gjovik, Norway | May 1994 | Yes |
| 5 | Leamington Spa, UK | October 1994 | Yes |
| 6 | Rome, Italy | April 1995 | Yes |
| 7 | Dresden, Germany | October 1995 | No |
| 8 | Tokyo, Japan | July 1996 | Yes |

Table F1.2. Annex 26 expert meetings held

## F6 Work carried out be modellers on the De Montfort Auditorium

Two institutes took an active part in modelling the ventilation system of the De Montfort Auditorium, using 2 and 3 dimensional CFD simulations. Dario Aiulfi of Sorane, Lausanne completed 2 D steady-state simulations (for more information see reference number 38, Appendix E).

Modelling of the existing natural ventilation system, a mechanical displacement system and a mechanical mixed ventilation system were carried out. Results of this work are presented in the Design Guide (editor: Dr J.R. Waters).

Frank Scholzen of ETH, Zurich carried out 3 D transient runs of the space. A paper, presented at the Roomvent conference in Yokohama, Japan, compares the CFD results obtained with the experimental results. Good agreement was not achieved due to the difficulties in representing accurate boundary conditions and in modelling heat transfer rates from the room surfaces. Finer grid resolution near boundaries would have enabled more accurate heat transfer coefficients to have been calculated. However this would have meant using a much larger number of cells thus considerably slowing down run times . See table F1.3 for a list of the work undertaken.

| Organisation | Researcher | Work carried out |
|---|---|---|
| Sorane, SA, Lausanne, Switzerland | Dario Aiulfi | Steady state 2 dimensional CFD analyses |
| ETH, Zurich, Switzerland | Alois Schalin<br>Frank Scholzen | Transient 3 dimensional CFD analyses (winter conditions)<br>Number of cells = 160,000 |

Table F1.3. CFD modelling performed on the De Montfort Auditorium

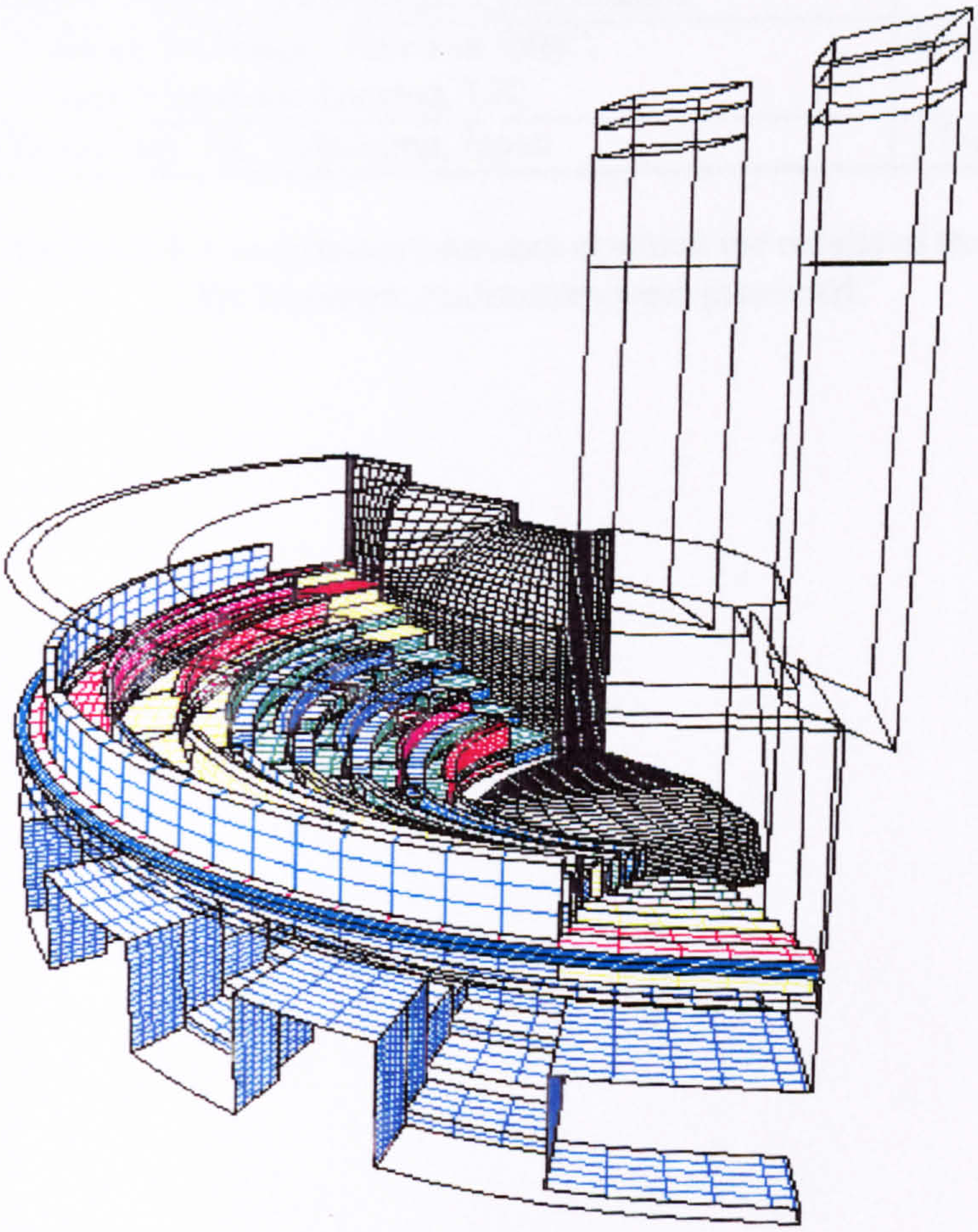

Figure F1 CFD grid of auditorium

## F7 Presentation of results

Results from the De Montfort measurement program were presented at the various annex meetings (see table F1.2). In addition, findings were presented at major conferences and seminars, some of which are listed in table F1.4.

| Conference/seminar | Date |
|---|---|
| European conference on energy performance and indoor climate in buildings, Lyon, France. | 24 to 26 November 1994 |
| "Passive buildings - future or folly". I Mech E seminar, London, UK. | 15 December 1994 |
| Roomvent '96, Yokohama, Japan | 17 to 19 July 1996 |

Table F1.4. Conferences/seminars at which the results of the measurement program for the De Montfort Auditorium were presented.

# Appendix G1: Calculation of internal temperatures (winter) Spreadsheet and formulae used

## Prediction of internal temperatures

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| to | 12 | | 15 | tsp | 19.5 | tmin | 12 |
| tao | 10 | tsi | 17 | TC1 | 70.269 | TC2 | 2.0278 |
| T(PH) | | 2 | | | 70.269 | | 2.0278 |
| Ts | | 7 | 18 | | | | |

| | 17 | 17 | 0 | 17 | 10 | |
|---|---|---|---|---|---|---|
| Time | | | Time | tai | tao | Htg Sys. |
| 0 | 17.00888 | 16.93963 | 0 | 16.93963 | 10 | 0 |
| 0.25 | 16.94872 | 16.87947 | 0.25 | 16.87947 | 10 | 0 |
| 0.5 | 16.88877 | 16.81952 | 0.5 | 16.81952 | 10 | 0 |
| 0.75 | 16.82904 | 16.75979 | 0.75 | 16.75979 | 10 | 0 |
| 1 | 16.76952 | 16.70026 | 1 | 16.70026 | 10 | 0 |
| 1.25 | 16.71021 | 16.64095 | 1.25 | 16.64095 | 10 | 0 |
| 1.5 | 16.65111 | 16.58186 | 1.5 | 16.58186 | 10 | 0 |
| 1.75 | 16.59222 | 16.52297 | 1.75 | 16.52297 | 10 | 0 |
| 2 | 16.53354 | 16.46429 | 2 | 16.46429 | 10 | 0 |
| 2.25 | 16.47507 | 16.40581 | 2.25 | 16.40581 | 10 | 0 |
| 2.5 | 16.4168 | 16.34755 | 2.5 | 16.34755 | 10 | 0 |
| 2.75 | 16.35875 | 16.28949 | 2.75 | 16.28949 | 10 | 0 |
| 3 | 16.30089 | 16.23164 | 3 | 16.23164 | 10 | 0 |
| 3.25 | 16.24325 | 16.174 | 3.25 | 16.174 | 10 | 0 |
| 3.5 | 16.18581 | 16.11655 | 3.5 | 16.11655 | 10 | 0 |
| 3.75 | 16.12857 | 16.05932 | 3.75 | 16.05932 | 10 | 0 |
| 4 | 16.07154 | 16.00228 | 4 | 16.00228 | 10 | 0 |
| 4.25 | 16.01471 | 15.94545 | 4.25 | 15.94545 | 10 | 0 |
| 4.5 | 15.95808 | 15.88882 | 4.5 | 15.88882 | 10 | 0 |
| 4.75 | 15.90165 | 15.8324 | 4.75 | 15.8324 | 10 | 0 |
| 5 | 15.84542 | 15.77617 | 5 | 15.77617 | 10 | 0 |
| 5.25 | 15.78939 | 15.72014 | 5.25 | 15.72014 | 10 | 0 |
| 5.5 | 15.73356 | 15.66431 | 5.5 | 15.66431 | 10 | 0 |
| 5.75 | 15.67793 | 15.60868 | 5.75 | 15.60868 | 10 | 0 |
| 6 | 15.6225 | 15.55325 | 6 | 15.55325 | 10 | 0 |
| 6.25 | 15.56726 | 15.49801 | 6.25 | 15.49801 | 10 | 0 |
| 6.5 | 15.51222 | 15.44297 | 6.5 | 15.44297 | 10 | 0 |
| 6.75 | 15.45738 | 15.38813 | 6.75 | 15.38813 | 10 | 0 |
| 7 | 15.40273 | 15.33348 | 7 | 15.33348 | 10 | 0 |
| 7.25 | 15.81675 | 15.27902 | 7.25 | 15.81675 | 10 | 100 |
| 7.5 | 16.24397 | 15.76058 | 7.5 | 16.24397 | 10 | 100 |
| 7.75 | 16.62163 | 16.18628 | 7.75 | 16.62163 | 10 | 100 |
| 8 | 16.95549 | 16.5626 | 8 | 16.95549 | 10 | 100 |
| 8.25 | 17.25063 | 16.89528 | 8.25 | 17.25063 | 10 | 100 |
| 8.5 | 17.51153 | 17.18936 | 8.5 | 17.51153 | 10 | 100 |
| 8.75 | 17.74217 | 17.44934 | 8.75 | 17.74217 | 10 | 100 |
| 9 | 17.94606 | 17.67916 | 9 | 17.94606 | 10 | 100 |
| 10 | 18.55101 | 18.36105 | 10 | 18.55101 | 10 | 100 |

## Prediction of internal temperatures

(winter)

| | | | | | | | |
|---|---|---|---|---|---|---|---|
| | | | | Dtsp | 2 | DTh | 2 |
| To | 12 | | 15 | Tsp | 21 | tmin | 12 |
| Tao | 6 | Tsi | 18 | TC1 | =4.6235*B5+24.034 | TC2 | =(-0.8296)*D5+16.131 |
| T(PH) | | 2 | | TC(Tai) | 35.5 | | =H5 |
| Ts | | 7 | 20 | TC(Tc) | 99.8 | | |
| | | | | | | | |
| | | =D5 | =D5 | 0 | =D5 | =$B$5 | |

| Time | Tai(1) | Tai(2) | Time | Tai(calc) | Tao | Htg Sys. |
|---|---|---|---|---|---|---|
| 0 | =IF(OR(B12<=$C$7,B12>$D$7),F9+($F$4-F9)*(1- EXP(-0.25/$F$6)),F9+($F$4-F9)*(1-EXP(-0.25/$H$6))) | =F9*EXP(-0.25/$F$6) | 0 | =IF(OR(B12<=$C$7,B12>$D | =$B$5 | =IF(F12<$H$4,100,0) |
| =B12+0.25 | =IF(OR(B13<=$C$7,B13>$D$7),F12+($F$4-F12)*(1- EXP(-0.25/$F$6)),F12+($F$4-F12)*(1-EXP(-0.25/$H$6))) | =F12*EXP(-0.25/$F$6) | =E12+0.25 | =IF(OR(B13<=$C$7,B13>$D | =$B$5 | =IF(F13<$H$4,100,0) |
| =B13+0.25 | =IF(OR(B14<=$C$7,B14>$D$7),F13+($F$4-F13)*(1- EXP(-0.25/$F$6)),F13+($F$4-F13)*(1-EXP(-0.25/$H$6))) | =F13*EXP(-0.25/$F$6) | =E13+0.25 | =IF(OR(B14<=$C$7,B14>$D | =$B$5 | =IF(F14<$H$4,100,0) |
| =B14+0.25 | =IF(OR(B15<=$C$7,B15>$D$7),F14+($F$4-F14)*(1- EXP(-0.25/$F$6)),F14+($F$4-F14)*(1-EXP(-0.25/$H$6))) | =F14*EXP(-0.25/$F$6) | =E14+0.25 | =IF(OR(B15<=$C$7,B15>$D | =$B$5 | =IF(F15<$H$4,100,0) |
| =B15+0.25 | =IF(OR(B16<=$C$7,B16>$D$7),F15+($F$4-F15)*(1- EXP(-0.25/$F$6)),F15+($F$4-F15)*(1-EXP(-0.25/$H$6))) | =F15*EXP(-0.25/$F$6) | =E15+0.25 | =IF(OR(B16<=$C$7,B16>$D | =$B$5 | =IF(F16<$H$4,100,0) |
| =B16+0.25 | =IF(OR(B17<=$C$7,B17>$D$7),F16+($F$4-F16)*(1- EXP(-0.25/$F$6)),F16+($F$4-F16)*(1-EXP(-0.25/$H$6))) | =F16*EXP(-0.25/$F$6) | =E16+0.25 | =IF(OR(B17<=$C$7,B17>$D | =$B$5 | =IF(F17<$H$4,100,0) |
| =B17+0.25 | =IF(OR(B18<=$C$7,B18>$D$7),F17+($F$4-F17)*(1- EXP(-0.25/$F$6)),F17+($F$4-F17)*(1-EXP(-0.25/$H$6))) | =F17*EXP(-0.25/$F$6) | =E17+0.25 | =IF(OR(B18<=$C$7,B18>$D | =$B$5 | =IF(F18<$H$4,100,0) |
| =B18+0.25 | =IF(OR(B19<=$C$7,B19>$D$7),F18+($F$4-F18)*(1- EXP(-0.25/$F$6)),F18+($F$4-F18)*(1-EXP(-0.25/$H$6))) | =F18*EXP(-0.25/$F$6) | =E18+0.25 | =IF(OR(B19<=$C$7,B19>$D | =$B$5 | =IF(F19<$H$4,100,0) |
| =B19+0.25 | =IF(OR(B20<=$C$7,B20>$D$7),F19+($F$4-F19)*(1- EXP(-0.25/$F$6)),F19+($F$4-F19)*(1-EXP(-0.25/$H$6))) | =F19*EXP(-0.25/$F$6) | =E19+0.25 | =IF(OR(B20<=$C$7,B20>$D | =$B$5 | =IF(F20<$H$4,100,0) |
| =B20+0.25 | =IF(OR(B21<=$C$7,B21>$D$7),F20+($F$4-F20)*(1- EXP(-0.25/$F$6)),F20+($F$4-F20)*(1-EXP(-0.25/$H$6))) | =F20*EXP(-0.25/$F$6) | =E20+0.25 | =IF(OR(B21<=$C$7,B21>$D | =$B$5 | =IF(F21<$H$4,100,0) |
| =B21+0.25 | =IF(OR(B22<=$C$7,B22>$D$7),F21+($F$4-F21)*(1- EXP(-0.25/$F$6)),F21+($F$4-F21)*(1-EXP(-0.25/$H$6))) | =F21*EXP(-0.25/$F$6) | =E21+0.25 | =IF(OR(B22<=$C$7,B22>$D | =$B$5 | =IF(F22<$H$4,100,0) |
| =B22+0.25 | =IF(OR(B23<=$C$7,B23>$D$7),F22+($F$4-F22)*(1- EXP(-0.25/$F$6)),F22+($F$4-F22)*(1-EXP(-0.25/$H$6))) | =F22*EXP(-0.25/$F$6) | =E22+0.25 | =IF(OR(B23<=$C$7,B23>$D | =$B$5 | =IF(F23<$H$4,100,0) |
| =B23+0.25 | =IF(OR(B24<=$C$7,B24>$D$7),F23+($F$4-F23)*(1- EXP(-0.25/$F$6)),F23+($F$4-F23)*(1-EXP(-0.25/$H$6))) | =F23*EXP(-0.25/$F$6) | =E23+0.25 | =IF(OR(B24<=$C$7,B24>$D | =$B$5 | =IF(F24<$H$4,100,0) |
| =B24+0.25 | =IF(OR(B25<=$C$7,B25>$D$7),F24+($F$4-F24)*(1- EXP(-0.25/$F$6)),F24+($F$4-F24)*(1-EXP(-0.25/$H$6))) | =F24*EXP(-0.25/$F$6) | =E24+0.25 | =IF(OR(B25<=$C$7,B25>$D | =$B$5 | =IF(F25<$H$4,100,0) |
| =B25+0.25 | =IF(OR(B26<=$C$7,B26>$D$7),F25+($F$4-F25)*(1- EXP(-0.25/$F$6)),F25+($F$4-F25)*(1-EXP(-0.25/$H$6))) | =F25*EXP(-0.25/$F$6) | =E25+0.25 | =IF(OR(B26<=$C$7,B26>$D | =$B$5 | =IF(F26<$H$4,100,0) |
| =B26+0.25 | =IF(OR(B27<=$C$7,B27>$D$7),F26+($F$4-F26)*(1- EXP(-0.25/$F$6)),F26+($F$4-F26)*(1-EXP(-0.25/$H$6))) | =F26*EXP(-0.25/$F$6) | =E26+0.25 | =IF(OR(B27<=$C$7,B27>$D | =$B$5 | =IF(F27<$H$4,100,0) |
| =B27+0.25 | =IF(OR(B28<=$C$7,B28>$D$7),F27+($F$4-F27)*(1- EXP(-0.25/$F$6)),F27+($F$4-F27)*(1-EXP(-0.25/$H$6))) | =F27*EXP(-0.25/$F$6) | =E27+0.25 | =IF(OR(B28<=$C$7,B28>$D | =$B$5 | =IF(F28<$H$4,100,0) |
| =B28+0.25 | =IF(OR(B29<=$C$7,B29>$D$7),F28+($F$4-F28)*(1- EXP(-0.25/$F$6)),F28+($F$4-F28)*(1-EXP(-0.25/$H$6))) | =F28*EXP(-0.25/$F$6) | =E28+0.25 | =IF(OR(B29<=$C$7,B29>$D | =$B$5 | =IF(F29<$H$4,100,0) |
| =B29+0.25 | =IF(OR(B30<=$C$7,B30>$D$7),F29+($F$4-F29)*(1- EXP(-0.25/$F$6)),F29+($F$4-F29)*(1-EXP(-0.25/$H$6))) | =F29*EXP(-0.25/$F$6) | =E29+0.25 | =IF(OR(B30<=$C$7,B30>$D | =$B$5 | =IF(F30<$H$4,100,0) |
| =B30+0.25 | =IF(OR(B31<=$C$7,B31>$D$7),F30+($F$4-F30)*(1- EXP(-0.25/$F$6)),F30+($F$4-F30)*(1-EXP(-0.25/$H$6))) | =F30*EXP(-0.25/$F$6) | =E30+0.25 | =IF(OR(B31<=$C$7,B31>$D | =$B$5 | =IF(F31<$H$4,100,0) |
| =B31+0.25 | =IF(OR(B32<=$C$7,B32>$D$7),F31+($F$4-F31)*(1- EXP(-0.25/$F$6)),F31+($F$4-F31)*(1-EXP(-0.25/$H$6))) | =F31*EXP(-0.25/$F$6) | =E31+0.25 | =IF(OR(B32<=$C$7,B32>$D | =$B$5 | =IF(F32<$H$4,100,0) |
| =B32+0.25 | =IF(OR(B33<=$C$7,B33>$D$7),F32+($F$4-F32)*(1- EXP(-0.25/$F$6)),F32+($F$4-F32)*(1-EXP(-0.25/$H$6))) | =F32*EXP(-0.25/$F$6) | =E32+0.25 | =IF(OR(B33<=$C$7,B33>$D | =$B$5 | =IF(F33<$H$4,100,0) |
| =B33+0.25 | =IF(OR(B34<=$C$7,B34>$D$7),F33+($F$4-F33)*(1- EXP(-0.25/$F$6)),F33+($F$4-F33)*(1-EXP(-0.25/$H$6))) | =F33*EXP(-0.25/$F$6) | =E33+0.25 | =IF(OR(B34<=$C$7,B34>$D | =$B$5 | =IF(F34<$H$4,100,0) |
| =B34+0.25 | =IF(OR(B35<=$C$7,B35>$D$7),F34+($F$4-F34)*(1- EXP(-0.25/$F$6)),F34+($F$4-F34)*(1-EXP(-0.25/$H$6))) | =F34*EXP(-0.25/$F$6) | =E34+0.25 | =IF(OR(B35<=$C$7,B35>$D | =$B$5 | =IF(F35<$H$4,100,0) |
| =B35+0.25 | =IF(OR(B36<=$C$7,B36>$D$7),F35+($F$4-F35)*(1- EXP(-0.25/$F$6)),F35+($F$4-F35)*(1-EXP(-0.25/$H$6))) | =F35*EXP(-0.25/$F$6) | =E35+0.25 | =IF(OR(B36<=$C$7,B36>$D | =$B$5 | =IF(F36<$H$4,100,0) |
| =B36+0.25 | =IF(OR(B37<=$C$7,B37>$D$7),F36+($F$4-F36)*(1- EXP(-0.25/$F$6)),F36+($F$4-F36)*(1-EXP(-0.25/$H$6))) | =F36*EXP(-0.25/$F$6) | =E36+0.25 | =IF(OR(B37<=$C$7,B37>$D | =$B$5 | =IF(F37<$H$4,100,0) |
| =B37+0.25 | =IF(OR(B38<=$C$7,B38>$D$7),F37+($F$4-F37)*(1- EXP(-0.25/$F$6)),F37+($F$4-F37)*(1-EXP(-0.25/$H$6))) | =F37*EXP(-0.25/$F$6) | =E37+0.25 | =IF(OR(B38<=$C$7,B38>$D | =$B$5 | =IF(F38<$H$4,100,0) |
| =B38+0.25 | =IF(OR(B39<=$C$7,B39>$D$7),F38+($F$4-F38)*(1- EXP(-0.25/$F$6)),F38+($F$4-F38)*(1-EXP(-0.25/$H$6))) | =F38*EXP(-0.25/$F$6) | =E38+0.25 | =IF(OR(B39<=$C$7,B39>$D | =$B$5 | =IF(F39<$H$4,100,0) |
| =B39+0.25 | =IF(OR(B40<=$C$7,B40>$D$7),F39+($F$4-F39)*(1- EXP(-0.25/$F$6)),F39+($F$4-F39)*(1-EXP(-0.25/$H$6))) | =F39*EXP(-0.25/$F$6) | =E39+0.25 | =IF(OR(B40<=$C$7,B40>$D | =$B$5 | =IF(F40<$F$4-$H$3,100,0) |
| =B40+0.25 | =IF(OR(B41<=$C$7,B41>$D$7),F40+($F$4-F40)*(1- EXP(-0.25/$F$6)),F40+($F$4-F40)*(1-EXP(-0.25/$H$6))) | =F40*EXP(-0.25/$F$6) | =E40+0.25 | =IF(OR(B41<=$C$7,B41>$D | =$B$5 | =IF(F41<$F$4-$H$3,100,0) |
| =B41+0.25 | =IF(OR(B42<=$C$7,B42>$D$7),F41+($F$4-F41)*(1- EXP(-0.25/$F$6)),F41+($F$4-F41)*(1-EXP(-0.25/$H$6))) | =F41*EXP(-0.25/$F$6) | =E41+0.25 | =IF(OR(B42<=$C$7,B42>$D | =$B$5 | =IF(F42<$F$4-$H$3,100,0) |
| =B42+0.25 | =IF(OR(B43<=$C$7,B43>$D$7),F42+($F$4-F42)*(1- EXP(-0.25/$F$6)),F42+($F$4-F42)*(1-EXP(-0.25/$H$6))) | =F42*EXP(-0.25/$F$6) | =E42+0.25 | =IF(OR(B43<=$C$7,B43>$D | =$B$5 | =IF(F43<$F$4-$H$3,100,0) |
| =B43+0.25 | =IF(OR(B44<=$C$7,B44>$D$7),F43+($F$4-F43)*(1- EXP(-0.25/$F$6)),F43+($F$4-F43)*(1-EXP(-0.25/$H$6))) | =F43*EXP(-0.25/$F$6) | =E43+0.25 | =IF(OR(B44<=$C$7,B44>$D | =$B$5 | =IF(F44<$F$4-$H$3,100,0) |
| =B44+0.25 | =IF(OR(B45<=$C$7,B45>$D$7),F44+($F$4-F44)*(1- EXP(-0.25/$F$6)),F44+($F$4-F44)*(1-EXP(-0.25/$H$6))) | =F44*EXP(-0.25/$F$6) | =E44+0.25 | =IF(OR(B45<=$C$7,B45>$D | =$B$5 | =IF(F45<$F$4-$H$3,100,0) |
| =B45+0.25 | =IF(OR(B46<=$C$7,B46>$D$7),F45+($F$4-F45)*(1- EXP(-0.25/$F$6)),F45+($F$4-F45)*(1-EXP(-0.25/$H$6))) | =F45*EXP(-0.25/$F$6) | =E45+0.25 | =IF(OR(B46<=$C$7,B46>$D | =$B$5 | =IF(F46<$F$4-$H$3,100,0) |
| =B46+0.25 | =IF(OR(B47<=$C$7,B47>$D$7),F46+($F$4-F46)*(1- EXP(-0.25/$F$6)),F46+($F$4-F46)*(1-EXP(-0.25/$H$6))) | =F46*EXP(-0.25/$F$6) | =E46+0.25 | =IF(OR(B47<=$C$7,B47>$D | =$B$5 | =IF(F47<$F$4-$H$3,100,0) |

# Appendix G2: Calculation of internal temperatures (summer) Spreadsheet and formulae used

## Response of 1st order system to sinusoidal input

| | | | | | | |
|---|---|---|---|---|---|---|
| Tmin | 16 | Tbase | 22 | | Timax | 23.0 |
| Tmax | 28 | Tvar | 6 | | Timin | 21.0 |
| | | | | | DTi | 2.1 |

| | | | |
|---|---|---|---|
| T(s) | $\frac{k}{s + a}$ | I(s) | $\frac{1}{s2 + w2}$ |
| Inc. | 1 | | |
| k | 1 | -k/2w2 | -7.30 |
| TC | 35 | | |
| a | 0.0286 | (a2-w2)/(a2+w2) | -0.976 |
| f0 | 0.03 | T0 | 33.33 |
| f0 | 0.0417 | T0 | 24 |
| w | 0.2618 | B | 7.123 |
| A | 1 | | |

| T (hour) | Sin(wt) | To(t) | Ti(t) | Cos(wt) | Theta |
|---|---|---|---|---|---|
| 0 | 0.0000 | 22.00 | 22.00 | -1.0000 | 0.0000 |
| 1 | -0.2588 | 20.45 | 22.27 | -0.9659 | 87.4549 |
| 2 | -0.5000 | 19.00 | 22.52 | -0.8660 | 88.8182 |
| 3 | -0.7071 | 17.76 | 22.73 | -0.7071 | 89.3176 |
| 4 | -0.8660 | 16.80 | 22.89 | -0.5000 | 89.6060 |
| 5 | -0.9659 | 16.20 | 23.00 | -0.2588 | 89.8171 |
| 6 | -1.0000 | 16.00 | 23.03 | 0.0000 | -90.0000 |
| 7 | -0.9659 | 16.20 | 23.00 | 0.2588 | -89.8171 |
| 8 | -0.8660 | 16.80 | 22.89 | 0.5000 | -89.6060 |
| 9 | -0.7071 | 17.76 | 22.73 | 0.7071 | -89.3176 |
| 10 | -0.5000 | 19.00 | 22.52 | 0.8660 | -88.8182 |
| 11 | -0.2588 | 20.45 | 22.27 | 0.9659 | -87.4549 |
| 12 | 0.0000 | 22.00 | 22.00 | 1.0000 | 0.0004 |
| 13 | 0.2588 | 23.55 | 21.73 | 0.9659 | 87.4549 |
| 14 | 0.5000 | 25.00 | 21.48 | 0.8660 | 88.8182 |
| 15 | 0.7071 | 26.24 | 21.27 | 0.7071 | 89.3176 |
| 16 | 0.8660 | 27.20 | 21.11 | 0.5000 | 89.6060 |
| 17 | 0.9659 | 27.80 | 21.00 | 0.2588 | 89.8171 |
| 18 | 1.0000 | 28.00 | 20.97 | 0.0000 | -90.0000 |
| 19 | 0.9659 | 27.80 | 21.00 | -0.2588 | -89.8171 |
| 20 | 0.8660 | 27.20 | 21.11 | -0.5000 | -89.6060 |
| 21 | 0.7071 | 26.24 | 21.27 | -0.7071 | -89.3176 |
| 22 | 0.5000 | 25.00 | 21.48 | -0.8660 | -88.8182 |
| 23 | 0.2588 | 23.55 | 21.73 | -0.9659 | -87.4549 |
| 24 | 0.0000 | 22.00 | 22.00 | -1.0000 | 0.0007 |

| | | | | | |
|---|---|---|---|---|---|
| | **Response of 1st order system to sinusoidal input** | | | | |
| Tmin | 16 | Tbase | =(B4+B5)/2 | | Timax |
| Tmax | 28 | Tvar | =(B5-B4)/2 | | Timin |
| | | | | | DTi |
| T(s) | k / s + a | | I(s) | 1 / s2 + w2 | |
| Inc. | 1 | | | | |
| k | 1 | -k/2w2 | | =-B12/(2*B17*B17) | |
| TC | 35 | | | | |
| a | =1/B13 | (a2-w2)/(a2+w2) | | =(B14*B14-B17*B17)/(B14*B14+B17*B1 | |
| f0 | =1/E15 | T0 | | =33.333333 | |
| f0 | =1/E16 | T0 | | 24 | |
| w | =2*3.14159265*$B$16 | B | | =E12*E14 | |
| A | 1 | | | | |
| T (hour) | Sin(wt) | To(t) | Ti(t) | Cos(wt) | Theta |
| 0 | =SIN((15*A22+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B22 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B22 | =COS((15*A22+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B22/($E$17+$E$12)/E22) |
| =A22+$B$11 | =SIN((15*A23+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B23 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B23 | =COS((15*A23+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B23/($E$17+$E$12)/E23)*180/3.14159 |
| =A23+$B$11 | =SIN((15*A24+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B24 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B24 | =COS((15*A24+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B24/($E$17+$E$12)/E24)*180/3.14159 |
| =A24+$B$11 | =SIN((15*A25+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B25 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B25 | =COS((15*A25+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B25/($E$17+$E$12)/E25)*180/3.14159 |
| =A25+$B$11 | =SIN((15*A26+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B26 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B26 | =COS((15*A26+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B26/($E$17+$E$12)/E26)*180/3.14159 |
| =A26+$B$11 | =SIN((15*A27+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B27 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B27 | =COS((15*A27+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B27/($E$17+$E$12)/E27)*180/3.14159 |
| =A27+$B$11 | =SIN((15*A28+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B28 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B28 | =COS((15*A28+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B28/($E$17+$E$12)/E28)*180/3.14159 |
| =A28+$B$11 | =SIN((15*A29+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B29 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B29 | =COS((15*A29+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B29/($E$17+$E$12)/E29)*180/3.14159 |
| =A29+$B$11 | =SIN((15*A30+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B30 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B30 | =COS((15*A30+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B30/($E$17+$E$12)/E30)*180/3.14159 |
| =A30+$B$11 | =SIN((15*A31+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B31 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B31 | =COS((15*A31+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B31/($E$17+$E$12)/E31)*180/3.14159 |
| =A31+$B$11 | =SIN((15*A32+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B32 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B32 | =COS((15*A32+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B32/($E$17+$E$12)/E32)*180/3.14159 |
| =A32+$B$11 | =SIN((15*A33+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B33 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B33 | =COS((15*A33+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B33/($E$17+$E$12)/E33)*180/3.14159 |
| =A33+$B$11 | =SIN((15*A34+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B34 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B34 | =COS((15*A34+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B34/($E$17+$E$12)/E34)*180/3.14159 |
| =A34+$B$11 | =SIN((15*A35+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B35 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B35 | =COS((15*A35+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B35/($E$17+$E$12)/E35)*180/3.14159 |
| =A35+$B$11 | =SIN((15*A36+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B36 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B36 | =COS((15*A36+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B36/($E$17+$E$12)/E36)*180/3.14159 |
| =A36+$B$11 | =SIN((15*A37+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B37 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B37 | =COS((15*A37+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B37/($E$17+$E$12)/E37)*180/3.14159 |
| =A37+$B$11 | =SIN((15*A38+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B38 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B38 | =COS((15*A38+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B38/($E$17+$E$12)/E38)*180/3.14159 |
| =A38+$B$11 | =SIN((15*A39+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B39 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B39 | =COS((15*A39+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B39/($E$17+$E$12)/E39)*180/3.14159 |
| =A39+$B$11 | =SIN((15*A40+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B40 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B40 | =COS((15*A40+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B40/($E$17+$E$12)/E40)*180/3.14159 |
| =A40+$B$11 | =SIN((15*A41+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B41 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B41 | =COS((15*A41+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B41/($E$17+$E$12)/E41)*180/3.14159 |
| =A41+$B$11 | =SIN((15*A42+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B42 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B42 | =COS((15*A42+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B42/($E$17+$E$12)/E42)*180/3.14159 |
| =A42+$B$11 | =SIN((15*A43+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B43 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B43 | =COS((15*A43+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B43/($E$17+$E$12)/E43)*180/3.14159 |
| =A43+$B$11 | =SIN((15*A44+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B44 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B44 | =COS((15*A44+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B44/($E$17+$E$12)/E44)*180/3.14159 |
| =A44+$B$11 | =SIN((15*A45+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B45 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B45 | =COS((15*A45+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B45/($E$17+$E$12)/E45)*180/3.14159 |
| =A45+$B$11 | =SIN((15*A46+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B46 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B46 | =COS((15*A46+180)*3.1415927/180) | =ATAN(($E$12-$E$17)*B46/($E$17+$E$12)/E46)*180/3.14159 |
| =A46+$B$11 | =SIN((15*A47+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B47 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B47 | | |
| =A47+$B$11 | =SIN((15*A48+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B48 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B48 | | |
| =A48+$B$11 | =SIN((15*A49+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B49 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B49 | | |
| =A49+$B$11 | =SIN((15*A50+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B50 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B50 | | |
| =A50+$B$11 | =SIN((15*A51+180)*3.1415927/180) | =$D$4+ $D$5*$B$18*B51 | =$D$4+$D$5*($E$12*$E$14 + $E$12)*B51 | | |

# Appendix G3: Calculation of room time constants Spreadsheet and formulae used

## Calculation of building time constants

| | | | | | | | | | |
|---|---|---|---|---|---|---|---|---|---|
| External wall area | | 60 | | his | =1/0.12 | | | | |
| Thickness | | 0.6 | | hso | =1/0.06 | | | | |
| Specific heat capacity | | 880 | | kw | 0.2 | | | | |
| Density | | 2000 | | | | | | | |
| | To | =0+273 | =5+273 | 10 | 15 | | | | |
| n (ach/h) | | 1 | =($C$15-D8)*$C$8/D8/($C$15-$C$8) | =($C$15-E8)*$C$8/E8/($C$15-$C$8) | =($C$15-F8)*$C$8/F8/($C$15-$C$8) | | | | |
| V | | 1000 | =SQRT(D9)*$C$9 | =SQRT(E9)*$C$9 | =SQRT(F9)*$C$9 | | | | |
| U value | | =1/(1/F4+1/F5+C5/F6) | | | | | | | |
| Internal wall area | | 500 | | Mass (o) | =C4*C5*C7 | mCp(o) | =F12*C6 | | |
| Thickness (inner w) | | 0.1 | | Mass (I) | =C7*C13*C12 | mCp(I) | =F13*C6 | | |
| | | | | | | | | | |
| Ti | | 20 | | Q/A | =C11*(C15-C16) | | | | |
| To | | 10 | | | | | | | |
| | | | | | | | | | |
| Tsi | | =C15-$F$15/$F$4 | | | | | | | |
| Tso | | =C16+$F$15/$F$5 | | | | | | | |
| Tsavg | | =(C18+C19)/2 | | | | | | | |
| | | | | | | | | | |
| | | | | | | | | TC (s) | TC(h) |
| To | Tsi | Tso | Tsavg | mCpDto | mCpDti | Sum | Heat loss | Sum/ht l | TC(h) |
| 0 | =$C$15-$F$15/$F$4 | =A25+$F$15/$F$5 | =(B25+C25)/2 | =$H$12*(D25-A25) | =$H$13*($C$15-A25) | =E25+F25 | =($C$11*$C$4+$C$9*$C$10/3)*($C$15-A25) | =G25/H25 | =I25/3600 |
| 5 | =$C$15-$F$15/$F$4 | =A26+$F$15/$F$5 | =(B26+C26)/2 | =$H$12*(D26-A26) | =$H$13*($C$15-A26) | =E26+F26 | =($C$11*$C$4+$D$9*$C$10/3)*($C$15-A26) | =G26/H26 | =I26/3600 |
| 10 | =$C$15-$F$15/$F$4 | =A27+$F$15/$F$5 | =(B27+C27)/2 | =$H$12*(D27-A27) | =$H$13*($C$15-A27) | =E27+F27 | =($C$11*$C$4+$E$9*$C$10/3)*($C$15-A27) | =G27/H27 | =I27/3600 |
| 15 | =$C$15-$F$15/$F$4 | =A28+$F$15/$F$5 | =(B28+C28)/2 | =$H$12*(D28-A28) | =$H$13*($C$15-A28) | =E28+F28 | =($C$11*$C$4+$F$9*$C$10/3)*($C$15-A28) | =G28/H28 | =I28/3600 |

# Appendix H: Derivation of formulae to predict internal conditions in summer within the auditorium

## Derivation of formula for calculating internal temperatures during "summer"

Assume a first order system (Laplace domain):

$$T(s) = \frac{K}{s+a}$$

where T(s) = system transfer function;

a = the inverse of the system time constant;

K = system gain

$$\frac{O(s)}{I(s)} = T(s)$$

$$O(s) = T(s)I(s)$$

where I(s) = input;

O(s) = output

Assume a sinusoidal variation in external temperature:

$$I(s) = \frac{\omega}{s^2 + \omega^2}$$

where $\omega$ = angular natural frequency (rad $s^{-1}$)

$$O(s) = \frac{K\omega}{(s+a)(s^2+\omega^2)}$$

$$O(s) = \frac{K_1}{(s+a)(s^2+\omega^2)}$$

$$O(s) = \frac{K_1}{(s+a)(s-j\omega)(s+j\omega)}$$

$$O(s) = \frac{A}{s+a} + \frac{B}{s-j\omega} + \frac{C}{s+j\omega}$$

$$K1 = A(s - j\omega)(s + j\omega) + B(s + a)(s + j\omega) + C(s + a)(s - j\omega)$$

$$K_1 = A(s^2 + \omega^2) + B(s^2 + ja\omega + j\omega s + as) + C(s^2 - ja\omega - j\omega s + as)$$

$$K_1 = s^2(A + B + C) + s( B(j\omega+a) + C(-j\omega+a)) + (A\omega^2 + Bja\omega - Cja\omega)$$

$$0 = A + B + C \qquad -A = B + C$$

$$0 = B(j\omega+a) + C(-j\omega+a)$$

$$K_1 = A\omega^2 + Bja\omega - Cja\omega$$

$$K_1 = -(B+C)\omega^2 + Bja\omega - Cja\omega$$

$$K_1 = B(ja\omega-\omega^2)-C(ja\omega+\omega^2)$$

$$\frac{K_1}{(ja\omega - \omega^2)} = B - C\frac{(ja\omega + \omega^2)}{(ja\omega - \omega^2)}$$

$$0 = B + C\frac{(-j\omega + a)}{(j\omega + a)}$$

$$\frac{K_1}{(ja\omega - \omega^2)} = -C\frac{(ja\omega + \omega^2)}{(ja\omega - \omega^2)} - C\frac{(-j\omega + a)}{(j\omega + a)}$$

$$K_1 = -C(ja\omega + \omega^2) - C\frac{(-j\omega + a)}{(j\omega + a)}(ja\omega - \omega^2)$$

$$K_1 = -C\left[(ja\omega + \omega^2) - \frac{(-j\omega + a)}{(j\omega + a)}(ja\omega - \omega^2)\right]$$

$$C = \frac{-K_1}{\left[(ja\omega + \omega^2) - \frac{(-j\omega + a)}{(j\omega + a)}(ja\omega - \omega^2)\right]}$$

$$C = \frac{-K_1(j\omega + a)}{\left[(ja\omega + \omega^2)(j\omega + a) + (j\omega - a)(ja\omega - \omega^2)\right]}$$

$$C = \frac{-K_1(j\omega + a)}{\left[-a\omega^2 + j\omega^2 + ja^2\omega + a\omega^2 - a\omega^2 + j\omega^3 - ja^2\omega + a\omega^2\right]}$$

$$C = \frac{-K_1(j\omega + a)}{2\omega^2(j\omega + a)}$$

$$C = \frac{-K_1}{2\omega^2}$$

$$B = -C\frac{(-j\omega + a)}{(j\omega + a)}$$

$$B = -\frac{-K_1}{2\omega^2}\frac{(-j\omega + a)}{(j\omega + a)}$$

$$\frac{(-j\omega + a)(a - j\omega)}{(j\omega + a)(a - j\omega)} = \frac{-ja\omega + a^2 - \omega^2 - ja\omega}{a^2 + \omega^2}$$

$$B = \left[\frac{a^2 - \omega^2 - j2a\omega}{a^2 + \omega^2}\right]\left[\frac{-K_1}{2\omega^2}\right]$$

$$A = -(B + C) = = \left[\frac{-K_1}{2\omega^2}\right]\left[\frac{a^2 - \omega^2 - j2a\omega}{a^2 + \omega^2} + 1\right]$$

We had:

$$O(s) = \frac{A}{s + a} + \frac{B}{s - j\omega} + \frac{C}{s + j\omega}$$

Therefore:

$o(t) = A\,e^{-at} + B\,e^{-j\omega t} + C\,e^{j\omega t}$

"A" can be ignored as $e^{-at}$ goes to zero as $t \rightarrow \infty$

$o(t) = B\,e^{-j\omega t} + C\,e^{j\omega t}$

$o(t) = B(\mathrm{Cos}(\omega t) - j\mathrm{Sin}(\omega t)) + C(\mathrm{Cos}(\omega t) + j\mathrm{Sin}(\omega t))$

Considering the real components of the above only:

$$o(t) = \left(\frac{-K_1}{2\omega^2}\right)\frac{(a^2 - \omega^2)}{(a^2 + \omega^2)}\mathrm{Cos}(\omega t) + \left(\frac{-K_1}{2\omega^2}\right)\mathrm{Cos}(\omega t)$$

Simplifying produces:

$$o(t) = \left(\frac{-K_1}{2\omega^2}\right)\left(\frac{(a^2 - \omega^2)}{(a^2 + \omega^2)} + 1\right)\mathrm{Cos}(\omega t)$$

**Phase angle (between external and internal temperature curves):**

$o(t) = B(\mathrm{Cos}(\omega t) - j\mathrm{Sin}(\omega t)) + C(\mathrm{Cos}(\omega t) + j\mathrm{Sin}(\omega t))$

Phase angle; $\theta$ = Imaginary component / real component

$$\theta = \tan^{-1}\left(\frac{C\mathrm{Sin}\omega t - B\mathrm{Sin}\omega t}{B\mathrm{Cos}\omega t + C\mathrm{Cos}\omega t}\right)$$